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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

# WARTIME REPORT

ORIGINALLY ISSUED

August 1944 as  
Advance Restricted Report 4H15

PERFORMANCE CHARACTERISTICS OF JOURNAL BEARINGS

WITH FORCED-FEED LUBRICATION

By S. A. McKee, H. S. White,  
A. D. Bell, and J. F. Swindells  
National Bureau of Standards



WASHINGTON

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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

ADVANCE RESTRICTED REPORT

PERFORMANCE CHARACTERISTICS OF JOURNAL BEARINGS

WITH FORCED-FEED LUBRICATION

By S. A. McKee, H. S. White,  
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SUMMARY

Friction and heat-dissipation data are presented for use in determining the load-carrying capacity of 2- by 1 $\frac{1}{4}$ -inch bearings with the following types of lining: copper-lead, lead-indium-coated silver, and lead-coated copper-lead. The tests were made in a four-bearing friction machine and covered operation with three clearances using three oils of different viscosity at three oil-inlet temperatures.

Values are computed to indicate the safe loads at various speeds for bearings lined with each of the three types of material. The data are presented in the form of charts convenient for obtaining relative safe-load values that may be used in estimating the effects of viscosity grade, viscosity index, oil-inlet temperature, bearing clearance, and bearing metal on the performance of aircraft-engine bearings in actual service.

The results indicate that both lead-indium-coated silver and lead-coated copper-lead bearings have greater load-carrying capacity than copper-lead bearings.

INTRODUCTION

Analysis of the action of a journal bearing indicates that from the standpoint of lubrication the performance of a bearing is dependent upon the generalized operating variable



$ZN/P$ , where  $Z$  is the absolute viscosity of the oil in the bearing,  $N$  is the speed of the journal, and  $P$  is the pressure on the projected area of the bearing. The performance of a bearing is also dependent upon its ability to dissipate heat. When the other factors remain constant, a rise in temperature in a bearing increases the rate of heat dissipation. The effect of a rise in temperature upon the rate of heat generation in a bearing, however, depends upon the conditions of operation. Bearing friction decreases with a rise in temperature at the high values of  $ZN/P$  and increases at low values. Consequently, if other factors are unchanged, the rate of heat generation also changes in the same manner. Accordingly, the bearing can reach a steady state of temperature distribution at high  $ZN/P$  values; hence this region of operation has been called the region of stable lubrication. At low values of  $ZN/P$  the increase in friction for a temperature rise becomes greater proportionally than the increase in the rate of heat dissipation by the bearing. Under these conditions the bearings cannot reach a steady state, hence the name "region of unstable lubrication."

This fundamental concept of a steady state for bearing operation is used as a basis for the work covered by this paper. The question of a steady state of temperature distribution has been discussed in a number of publications. (See reference 1.) An analytical treatment pertinent to the work in this paper is given in reference 2. Friction and heat-dissipation data were obtained over a wide range of conditions involving changes in operational and design factors. Data were also obtained at low  $ZN/P$  values pertinent to the determination of the limiting value of  $ZN/P$  for safe operation with each of three types of bearing materials. From these data computations of load-carrying capacity are made and are presented in the form of charts convenient for use in quantitative estimates of bearing performance. Thus the relative effects of changes in the various operational and design factors on the performance of bearings of other sizes in actual service can be evaluated.

These tests are part of a research program on lubrication of aircraft-engine bearings being carried out by the National Bureau of Standards with the financial assistance of the National Advisory Committee for Aeronautics.



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## APPARATUS

## Four-Bearing Friction Machine

Photographs of the four-bearing friction machine and parts used in this investigation are given in figures 1 and 2. The machine consists essentially of four test bearings mounted on a common shaft. The bearings used in these tests were solid steel sleeves lined with thin layers of the bearing metals. They are mounted in self-aligning ball-bearing swivels which are prevented from rotating by flat spring torque absorbers. The two outside bearings are mounted directly in the ends of the bearing housing. The two inner bearings are mounted in plates sliding in guides. Load is applied equally to the two inner bearings from a hydraulic jack at the base of the housing through two jack plungers. The reaction from this load is taken by the two outer bearings and; since the bearings are symmetrically spaced, all are equally loaded.

The complete housing floats on the horizontal test shaft and hence acts as a cradle dynamometer. The frictional torque is measured by a direct-reading torque indicator contacting a torque arm mounted on the housing. The torque indicator used in most of these tests consists of a small diaphragm jack and pressure gage. In later tests this indicator was replaced by a dynamometer scale. For torque measurements greater than the capacity of indicator or scale, weights are added to a pan suspended from the torque arm. Stops are provided to limit the motion of the torque arm.

The use of a flexible diaphragm (synthetic rubber) in the hydraulic jack prevents leakage. This permits the use of a hand-operated injector which can be set to maintain a constant load. An automatic load release in the hydraulic loading system releases the pressure in the jack when the frictional torque gets too high under conditions where the bearings are approaching seizure.

Oil is fed to the bearings through the drilled test shaft with two oil holes at the axial center of each bearing. Oil-feed pressure is maintained by a motor-driven gear pump with a relief valve for pressure control.

Bearing temperatures are measured by thermocouples mounted on the loaded sides of the bearings.



## Shafts and Bearings

The shafts used in these tests were made of crankshaft steel conforming to Pratt & Whitney Aircraft specification 190.

Three types of bearing material were used. Bearing sets 1a, 1b, and 1c had copper-lead linings conforming to Pratt & Whitney Aircraft specification 121. Set 2a had silver linings coated with a 0.0015-inch thickness of lead which was treated with indium, and set 3a was lined with lead-coated copper-lead.

The essential dimensions of bearings and shaft are as follows:

Bearing set	1a	1b	1c	2a	3a
Length of bearings, L, in.	1.275	1.275	1.275	1.275	1.275
Av. diam. of bearings, in.	2.0574	2.0593	2.0616	2.0585	2.0583
Av. diam. of shaft at journals, D, in.	2.0556	2.0556	2.0556	2.0542	2.0542
Av. clearance, C, in.	0.0018	0.0037	0.0060	0.0043	0.0041
Clearance diam. ratio, C/D	0.0009	0.0018	0.0029	0.0021	0.0020
Length diam. ratio, L/D	0.620	0.620	0.620	0.621	0.621

## Lubricants

The lubricants used were a Pennsylvania oil (NBS laboratory reference No. J3-120), a Navy contract 1080 oil, and an SAE 20 motor oil. The viscosity data for these oils are as follows:

Lubricant	Viscosity			
	Saybolt Universal seconds		Centistokes	
	100° F	210° F	100° F	210° F
NBS reference J3-120	1766	124.4	382.3	26.04
Navy contract 1080	781	77.5	169.1	14.95
SAE 20	371	58.6	80.2	9.82



## TEST RUNS

## Heat-Dissipation Tests

In the heat-dissipation tests the bearings were operated in the region of stable lubrication at the higher values of ZN/P. Test runs were made at constant speeds and at a number of constant loads which were successively increased during each test run. The data were obtained with the apparatus thoroughly "warmed up" and with the bearings in a steady state of temperature distribution. The range of conditions covered in the heat-dissipation tests with each set of bearings is as follows:

## OPERATING CONDITIONS FOR HEAT-DISSIPATION TESTS

Bearing set	1a	1b	1c	2a	3a
Oil used	J3-120 1080 SAE 20	J3-120 1080 SAE 20	J3-120 1080 SAE 20	J3-120 1080 SAE 20	J3-120 ----- -----
Speed, N, rpm	2000 and 3000	2000 and 3000	2000 and 3000	2000 and 3000	2000 and 3000
Pressure on projected area of bearings, P, lb/sq in.	725 to 2289	725 to 4057	725 to 4332	725 to 2970	725 to 4332
Av. oil-feed pressure p, lb/sq in.	37	35	34	35	36
Oil-inlet temperature, °F	150 200 250	150 200 250	150 200 250	150 200 250	150 --- 250
Bearing temperature, °F	224 to 343	184 to 322	170 to 297	174 to 306	203 to 334
Case temperature, °F	174 to 247	155 to 260	146 to 253	158 to 253	167 to 263
Av. room temperature, °F	85	79	79	84	82



Some tests also were made at various oil-feed pressures using the Navy contract 1080 oil at 200° F oil-inlet temperature and a journal speed of 2000 rpm. These tests include operation with bearing set 1a at oil-feed pressures of 37 and 103 pounds per square inch, set 1b at 15, 35, 57, and 103 pounds per square inch, and set 1c at 15, 34, and 97 pounds per square inch.

### Friction Tests at Low ZN/P

In the tests to determine the effects of the bearing material upon the frictional characteristics of the bearings, particular attention was given to operation at low values of ZN/P in the region of unstable lubrication. These tests were made on bearing sets 1b, 2a, and 3a - all of which had approximately the same C/D ratios. The test runs were made at a constant speed of 2000 rpm and at a number of loads which were successively increased during each test run. All the tests were run with the same oil, the same oil-inlet temperature, and about the same oil-feed pressure. The entire apparatus was thoroughly warmed up before each test. The tests were made with the bearings in a "run-in" condition. Three consecutive runs covering the complete load range were made with each set of bearings. The operating conditions were as follows:

### OPERATING CONDITIONS FOR TESTS AT LOW ZN/P

Bearing set	1b	2a	3a
Oil used	J3-120	J3-120	J3-120
Speed, N, rpm	2000	2000	2000
Pressure on projected area of bearings, P, lb/sq in.	971 to 5144	971 to 7055	971 to 7055
Av. oil-feed pressure p, lb/sq in.	34	32	33
Oil-inlet temperature, °F	250	250	250
Bearing temperature, °F	276 to 317	276 to 349	276 to 360
Case temperature, °F	218 to 248	223 to 272	216 to 266
Av. room temperature, °F	82	82	84



## DISCUSSION OF RESULTS

## Heat-Dissipation Characteristics

The data obtained in the heat-dissipation tests with bearing set 1a (copper-lead,  $C/D = 0.0009$ ) are given in figure 3. These data cover operation with the three grades of oil at three oil-inlet temperatures, at various loads, and at two journal speeds. In this figure  $H'$ , the rate of heat dissipation for the four bearings in inch-pounds per minute (used instead of the conventional Btu/min), is plotted against  $\Delta T_a$ , the average rise in temperature in degrees Fahrenheit of the bearings above the ambient. These tests were run under conditions of a steady state of temperature distribution; hence the values of  $H'$  were obtained from observations of frictional torque.

Similar data for bearing sets 1b (copper-lead,  $C/D = 0.0018$ ), 1c (copper-lead,  $C/D = 0.0029$ ), and 2a (lead-indium-coated silver,  $C/D = 0.0021$ ) are shown in figures 4, 5, and 6, respectively. The tests with set 3a (lead-coated copper-lead,  $C/D = 0.0020$ ) were confined to runs with the J3-120 oil at oil-inlet temperatures of 150° F and 250° F. These data are shown in figure 7.

The heat-dissipation data for sets 1a, 1b, and 1c when operating at 2000 rpm using the Navy contract 1080 oil at 200° F oil-inlet temperature and at various oil-feed pressures are given in figure 8. In this figure the bearing set and the oil-feed pressure are indicated for each curve.

The rate of heat removal by the oil flowing through the machine for typical conditions with bearing sets 1a, 1b, and 1c are shown by the dotted curves in figure 9. The curves were obtained from data on the rate of flow of the oil, the temperature difference between the oil entering the end of the shaft and the oil leaving the case, and the specific heat of the particular oil used. The solid curves in this figure represent the total rate of heat dissipation as determined by the torque measurements.

Analysis of the data given in figures 3 to 9 indicates that with forced-feed lubrication under the conditions covered by these tests much of the heat generated in the bearings is carried away by the oil; hence their operating temperatures are influenced greatly by the temperature of the entering oil and factors affecting oil flow.



The general characteristics for a given temperature rise above the ambient are as follows:

(a) The rate of heat dissipation increases with a decrease in viscosity grade (indicated in figs. 3 to 6).

(b) The rate of heat dissipation increases with a decrease in oil-inlet temperature (indicated in figs. 3 to 7).

(c) The rate of heat dissipation increases with an increase in oil-feed pressure (indicated in fig. 8).

(d) The rate of heat dissipation increases with an increase in clearance (indicated by comparison of curves for same oil and oil-inlet temperature in figs. 3, 4, and 5).

The effect of different bearing metals upon the heat-dissipation characteristics of the bearings is relatively small (indicated by a comparison of curves in figs. 4, 6, and 7). Some of the differences shown can be attributed to the differences in clearance between the three sets of bearings. Graphical interpolation of the data given in figures 3, 4, and 5 provides a means for correcting the curves in figures 6 and 7 so that they are representative of the same clearance (0.0037 in.) as the curves in figure 4. The corrected curves indicate that under the same conditions the rate of heat dissipation of the copper-lead bearings and the lead-coated copper-lead bearings was about the same and slightly less than of the lead-indium-coated silver bearings.

### Frictional Characteristics in the Region of

#### Unstable Lubrication

The data obtained in the tests of the copper-lead bearings (set 1b,  $C/D = 0.0018$ ), the lead-indium-coated silver bearings (set 2a,  $C/D = 0.0021$ ), and the lead-coated copper-lead bearings (set 5a,  $C/D = 0.0020$ ) when operating at the lower values of  $ZN/P$  in the region of unstable lubrication are shown in figure 10, where  $f$ , the coefficient of friction, is plotted against  $ZN/P$ , where  $Z$  is in centipoises,  $N$  in revolutions per minute, and  $P$  in pounds per square inch.

One of the chief difficulties in obtaining representative data in the region of unstable lubrication is that in this region of operation, frequently the frictional characteristics of a bearing are not constant. This is especially



true when operating at the high loads, speeds, and operating temperatures used in these tests. Under some conditions the bearing is improved with operation while at others it is impaired. This was found to be especially characteristic of the copper-lead bearings. From this consideration, the curves in figure 10 were chosen as being reasonably representative of the average performance of the respective bearings (in the run-in condition) under the test conditions. In these test runs, precautions were taken to control, as far as possible, all variables except the bearing metal. All three sets of bearings had approximately the same clearance and were run at the same speed with the same oil at the same oil-inlet temperature and about the same oil-feed pressure.

The friction data obtained in these tests are shown also in figure 11. In this figure the values of  $f$  obtained at a given load with a given set of bearings are plotted against  $P$ , the pressure on the projected area of the bearing. These curves provide an indication of the load-carrying capacity of the bearings, since the load at minimum  $f$  represents approximately the maximum load at which the respective bearings will operate in the region of stable lubrication under the given conditions with the given lubricant.

The data shown in figures 10 and 11 indicate that the lead-indium-coated silver bearings had a relatively higher friction. However, the load-carrying capacity of these bearings was about equal to that of the lead-coated copper-lead bearings, and markedly higher than that of the copper-lead bearings.

#### COMPUTATIONS OF LOAD-CARRYING CAPACITY

The data in figure 11 provide an indication of the safe loads for the three sets of bearings tested when operating at the given speed using the particular oil, oil-inlet temperature, and oil-feed pressure. The computed load-carrying capacities of all five sets of bearings when operating at different speeds using different oils, oil-inlet temperatures, and oil-feed pressures are given in figures 12 to 17. (The method used in making these safe-load computations is described in detail in reference 2.) Each curve in these figures represents the limits of safe pressure  $P$  for the various values of the speed  $N$  when using the designated bearings, oil, oil-inlet temperature, and oil-feed pressure.



The computations are based on the heat-dissipation data given in figures 3 to 8, the viscosity-temperature characteristics of the oils used, the ambient temperature, and the limits of safe operation as indicated by the point of minimum  $f$  in the curve of  $f$  plotted against  $ZN/P$  for the bearings.

The limits for safe operation used in these computations were obtained from the curves in figure 10. The values used for the copper-lead bearings are  $ZN/P = 3.0$  and minimum  $f = 0.00135$ ; for the lead-indium-coated silver bearings  $ZN/P = 1.7$  and minimum  $f = 0.00160$ ; and for the lead-coated copper-lead bearings  $ZN/P = 1.7$  and minimum  $f = 0.00140$ . The computations for the lead-indium-coated silver bearings are based upon the heat-dissipation-temperature relations shown in figure 6 with the corrections (previously mentioned) applied so that they are representative of bearings having a clearance of 0.0037 inch (clearance-diameter ratio = 0.0018). The heat-dissipation curves for the lead-coated copper-lead bearings when corrected for clearance were practically the same as corresponding curves for the copper-lead bearings; hence the safe-load computations for the lead-coated copper-lead bearings are based on the heat-dissipation curves in figure 4. Computations for all sets of bearings were based on an assumed ambient temperature of 86° F.

Analysis of the curves given in figures 12 to 17 indicates that, all other factors remaining unchanged, the load-carrying capacity of the bearings is increased by an increase in viscosity grade, is decreased by an increase in oil-inlet temperature, is increased by an increase in oil-feed pressure, and is increased by an increase in clearance. Comparison of the curves for the bearings of different metals indicates that over the range of conditions covered the lead-indium-coated silver and the lead-coated copper-lead bearings are about equal in load-carrying capacity and both markedly superior to the copper-lead bearings in this respect.

#### SUMMARY OF RESULTS

Data are presented showing the heat-dissipation characteristics of 2- by 1 $\frac{1}{4}$ -inch bearings operating in a four-bearing friction machine with forced-feed lubrication at two speeds over a wide range of loads. These data show the influence of such factors as viscosity grade, oil-inlet temperature, oil-feed pressure, bearing clearance, and bearing



metal. Friction data are given which provide a comparison of the load-carrying capacity of 2- by  $1\frac{1}{4}$ -inch bearings lined with copper-lead, lead-indium-coated silver, and lead-coated copper-lead when each type was operated at the same speed with the same oil and oil-inlet temperature. From these data values of safe loads at various speeds are computed. These data show the effects of changes in the various factors upon the load-carrying capacity of the bearings.

Analysis of these data indicates the following effects of the individual variables (all others remaining constant) upon the bearing temperature and safe load for a bearing operating with forced-feed lubrication at a constant speed:

Change	Effect on bearing temperature <sup>1</sup>	Effect on computed safe pressure on projected area
Increase pressure on projected area	Increase	-----
Increase viscosity grade <sup>2</sup>	Increase	Increase
Increase oil-inlet temperature	Increase	Decrease
Increase oil-feed pressure	Decrease	Increase
Increase clearance-diameter ratio	Decrease	Increase
Type of bearing metal	Negligibly small	Lead-indium silver and lead-coated copper-lead about the same; copper-lead markedly lower.

<sup>1</sup>Based on data obtained when operating in the region of stable lubrication.

<sup>2</sup>Involves changing to an oil of higher viscosity at all operating temperatures.



## APPENDIX

## SAFE-LOAD CHARTS

In considering these data from the standpoint of application to bearings under actual operating conditions, it is of interest to make a comparison of the two cases. The materials used in the shafts and bearings were the same as those used in aircraft engines. The shaft diameter was smaller than most engine crankshafts but it was of the same order of magnitude. The length-diameter ratio was within the range of engine bearings and the clearance-diameter ratios covered the same range. The lubricants were typical for aviation engine oils and their viscosities covered ordinary usage. The method for feeding the oil to the bearings (oil holes in shaft), the oil-feed pressures, and the oil-inlet temperatures also were typical of engine operation.

The major differences are in the types of loading and in the sources of heat. In the engine the loads vary in intensity and direction relative to the bearings while in the laboratory machine the loads are constant both in intensity and direction relative to the bearings. In the engine the operating temperatures are influenced by heat from the cylinders; hence from this standpoint, conditions may be more severe in the engine. However, the temperatures observed with the laboratory machine cover the range of probable engine bearing operating temperatures. The engine bearings are subject to impact and vibrational loads which are relatively small in the laboratory machine. In one respect, however, conditions in the laboratory machine may be more severe. In the laboratory machine one portion of the bearing is constantly under high stress, while in the engine a given portion of the bearing is under high stress only at certain positions in a cycle. Also, in the laboratory machine an oil film is maintained in the bearings only by the mechanism of the wedging action, while in the engine the load tends to force the shaft and bearing together at various points where the surfaces are already well coated with oil. Other factors, not taken into account by the laboratory machine, are the fatigue characteristics of the bearing metals, misalignment and distortion, and contamination of the oil.

Because of these differences it would not be expected that the numerical results obtained in the laboratory machine would be directly applicable to full-scale engine operation.



On the other hand, relative differences shown by the laboratory data should be of significance to performance in service and when used in conjunction with service experience provide useful guides in practical problems relating to aircraft engine bearings. Accordingly the safe-load data are given in chart form convenient for this purpose.

#### Charts Showing Safe Loads for Army and Navy Specification Oils at Various Oil-Inlet Temperatures

Analysis of the data given in figures 12 to 16 indicates that for a given bearing operating at a given speed the safe load at a given oil-inlet temperature is a function of the viscosity of the oil at that temperature. Also when a given oil is used, the safe load is approximately a linear function of the oil-inlet temperature. From these relations the safe-load chart shown in figure 18 was developed for a 2- by  $1\frac{1}{4}$ -inch copper-lead bearing having a clearance-diameter ratio of 0.0018 (which approximates the clearance-diameter ratios of typical aircraft engine bearings). In this figure the maximum safe pressures (plotted as ordinates) are shown for this bearing when operating at 2500 rpm using the designated grades of Army and Navy specification aviation oils at the various oil-inlet temperatures (plotted as abscissas). The upper line for a designated grade represents the upper viscosity limit of that grade, while the lower line represents the lower viscosity limit. Both limits are based on a viscosity index of 100.

Similar charts for the operation of lead-indium-coated silver and lead-coated copper-lead bearings operating at 2500 rpm are shown in figures 19 and 20, respectively.

While the absolute values shown in these figures are applicable only to the particular conditions upon which the load-carrying capacity data are based, the charts provide a means for estimating the relative effects of changes in viscosity grade and oil-inlet temperature on the load-carrying capacity of aircraft engine bearings in actual service.

One example of a practical use for such a chart is given by the hypothetical case which follows:

Aircraft engine, normally operating at 2500 rpm using copper-lead bearings and Navy symbol 1100 oil. Service records indicate that there is little bearing trouble if oil-



inlet temperature is maintained below  $220^{\circ}$  F. Horsepower of the engine is to be increased making the load on the bearings 15 percent greater. What grade of oil should be used and what is the limiting oil-inlet temperature?

From figure 18 it will be noted that the mean value of the safe load for a Navy symbol 1100 oil at  $220^{\circ}$  F oil-inlet temperature is about 4100 pounds per square inch. The new requirements are a 15-percent increase or about 4700 pounds per square inch. Again referring to figure 18 it is seen that this load is approximately a mean value for a Navy symbol 1100 oil at  $165^{\circ}$  F oil-inlet temperature and for a Navy symbol 1120 oil at  $190^{\circ}$  F oil-inlet temperature. The choice as to which of the oils could be used to better advantage would depend upon other factors such as cylinder wall and ring lubrication and oil-cooling capacity. While the final answer to this problem could be obtained only by service performance, these estimates should be useful as a guide in making the changes necessary.

Charts for Determining Safe Loads for Oils of  
Different Viscosity Grade and Viscosity Index  
at Various Oil-Inlet Temperatures

The charts given in figures 21, 22, and 23 are for general use with oils covering a wide range of viscosity index. These charts are based on the same relations as those in figures 18, 19, and 20 and pertain to the same bearings (2- by  $1\frac{1}{2}$ -inch copper-lead, lead-indium-coated silver, and lead-coated copper-lead, respectively) operating at the same speed, 2500 rpm. In these charts the safe pressures are plotted as ordinates and the oil-inlet temperatures as abscissas. At the left there is a composite scale consisting of four vertical scales for viscosity at  $100^{\circ}$  F in Saybolt seconds laid out on a horizontal scale for viscosity index (computed from relation of change in viscosity at given bearing temperatures with change in viscosity index). A similar scale for Saybolt viscosities at  $210^{\circ}$  F is laid out at the right.

To use these charts, plot on the composite scale at the left of the chart a point representing the viscosity of the oil at  $100^{\circ}$  F and the viscosity index of the oil. Plot a point representing the viscosity of the oil at  $210^{\circ}$  F and its viscosity index on the scale at the right of the chart. Draw a straight line between these two points. This line



represents approximately the limits of safe pressure for this oil at the various oil-inlet temperatures. These charts are convenient for estimating the effects of changes in viscosity index on the load-carrying capacity of the bearings. This is illustrated in figure 21, where lines are plotted for two oils having the same viscosity at  $210^{\circ}\text{F}$  (100 sec) but with different viscosity indices, 90 and 110, respectively. From these lines it will be noted that under the conditions covered an increase in viscosity index from 90 to 110 provided an increase in load-carrying capacity of about 4 percent at  $150^{\circ}\text{F}$  oil-inlet temperature to about 7 percent at  $250^{\circ}\text{F}$  oil-inlet temperature.

Charts for Determining Safe Loads for Bearings of  
Different Clearance Using Oils of Various  
Viscosity Grade at Various Oil-Inlet Temperatures

A third type of chart may be used to estimate the effects of changes in clearance when using various grades of oil at various oil-inlet temperatures. A chart of this type for use with 2- by  $1\frac{1}{4}$ -inch copper-lead bearings operating at 2500 rpm is shown in figure 24. In this figure the safe-load scale is along the left border and one for clearance-diameter ratios along the lower border. The right side of the chart contains a composite scale consisting of three vertical scales for kinematic viscosity laid out at  $150^{\circ}\text{F}$ ,  $200^{\circ}\text{F}$ , and  $250^{\circ}\text{F}$ , respectively, on a horizontal scale for oil-inlet temperature. These scales apply to oils of approximately 100 viscosity index. The shaded areas in this composite scale represent the various grades of Army and Navy specification oils of 100 viscosity index. The numbers in each area indicate the respective grade.

In using this chart the viscosity of the oil at the particular oil-inlet temperature under consideration is plotted on the composite scale at the right. A straight line is then drawn between this point and the bull's-eye located near the lower left-hand corner of the chart. This straight line will define approximately the limiting safe pressures for the clearance-diameter ratios indicated along the lower border when using the respective oil, oil-inlet temperature, and speed.

Such a line for a Navy symbol 1100 oil (mean value) at  $200^{\circ}\text{F}$  oil-inlet temperature is shown in figure 24. This



line may be used to indicate the effects of machining tolerances on the performance of the bearings. For example, consider a 2-inch bearing where the machining tolerances are such that the clearance may vary from 0.003 inch to 0.005 inch (clearance-diameter ratios of 0.0015 to 0.0025). Under the conditions represented by the line in figure 24, the safe load at the large clearance is about 5000 pounds per square inch while with the small clearance the safe load is about 3800 pounds per square inch, which represents a reduction of about 24 percent. While these particular values are applicable only to the conditions represented by the line in figure 22, changes of this magnitude would be expected with service bearings for corresponding changes in clearance-diameter ratio.

In this connection it should be noted that these data are based solely upon the effects of  $C/D$  ratio upon the heat dissipation of the bearing and do not take into account the effect of  $C/D$  ratio upon the point of minimum  $f$  in the curve of  $f$  plotted against  $ZN/P$ . Theoretical hydrodynamic equations for bearings with no end leakage indicate that the  $C/D$  ratio affects the minimum film thickness and hence the load-carrying capacity of the bearings. During the course of this investigation friction data at low  $ZN/P$  were obtained with bearings of different  $C/D$  ratio. These data, however, differ markedly from the theoretical equations. The location of the point of minimum  $f$  varied somewhat with different sets of bearings, but any trend with change in  $C/D$  ratio was sufficiently small to be blanketed by the effects of differences in surface conditions or other factors. Further work is necessary to evaluate the effects of  $C/D$  ratio at low  $ZN/P$ .

Similar charts for use with 2- by  $1\frac{1}{2}$ -inch lead-indium-coated silver and lead-coated copper-lead bearings operating at 2500 rpm are given in figures 25 and 26, respectively.

#### REFERENCES

1. Hersey, Mayo Dyer: Theory of Lubrication. John Wiley & Sons, Inc., 1936.
2. McKee, Samuel A.: Friction and Temperature as Criteria for Safe Operation of Journal Bearings. Res. Paper RP1295, Bur. Standards Jour. Res., vol. 24, no. 5, May 1940, pp. 491-508.



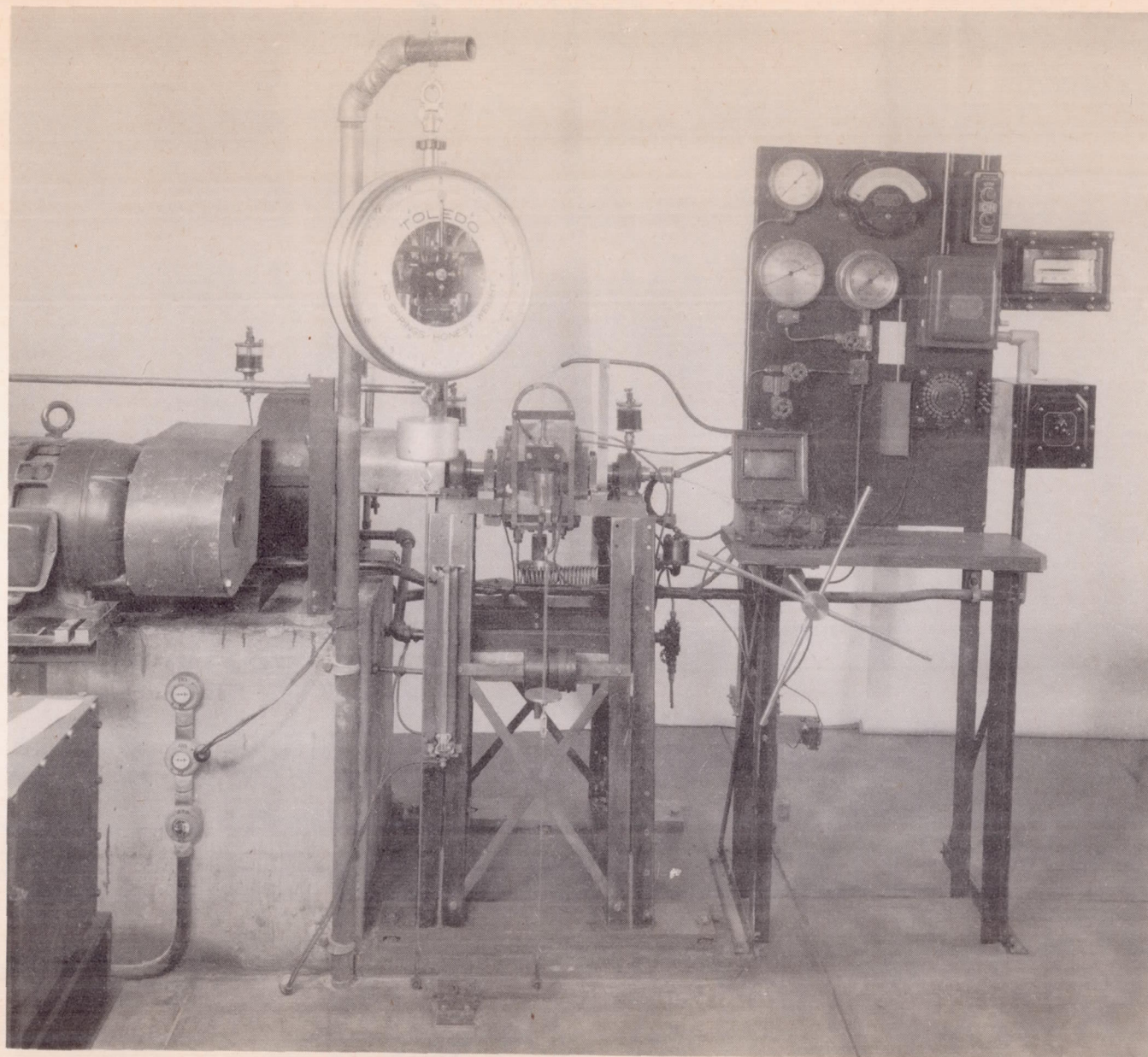


Fig. 1

Figure 1.- Photograph of friction machine installation.



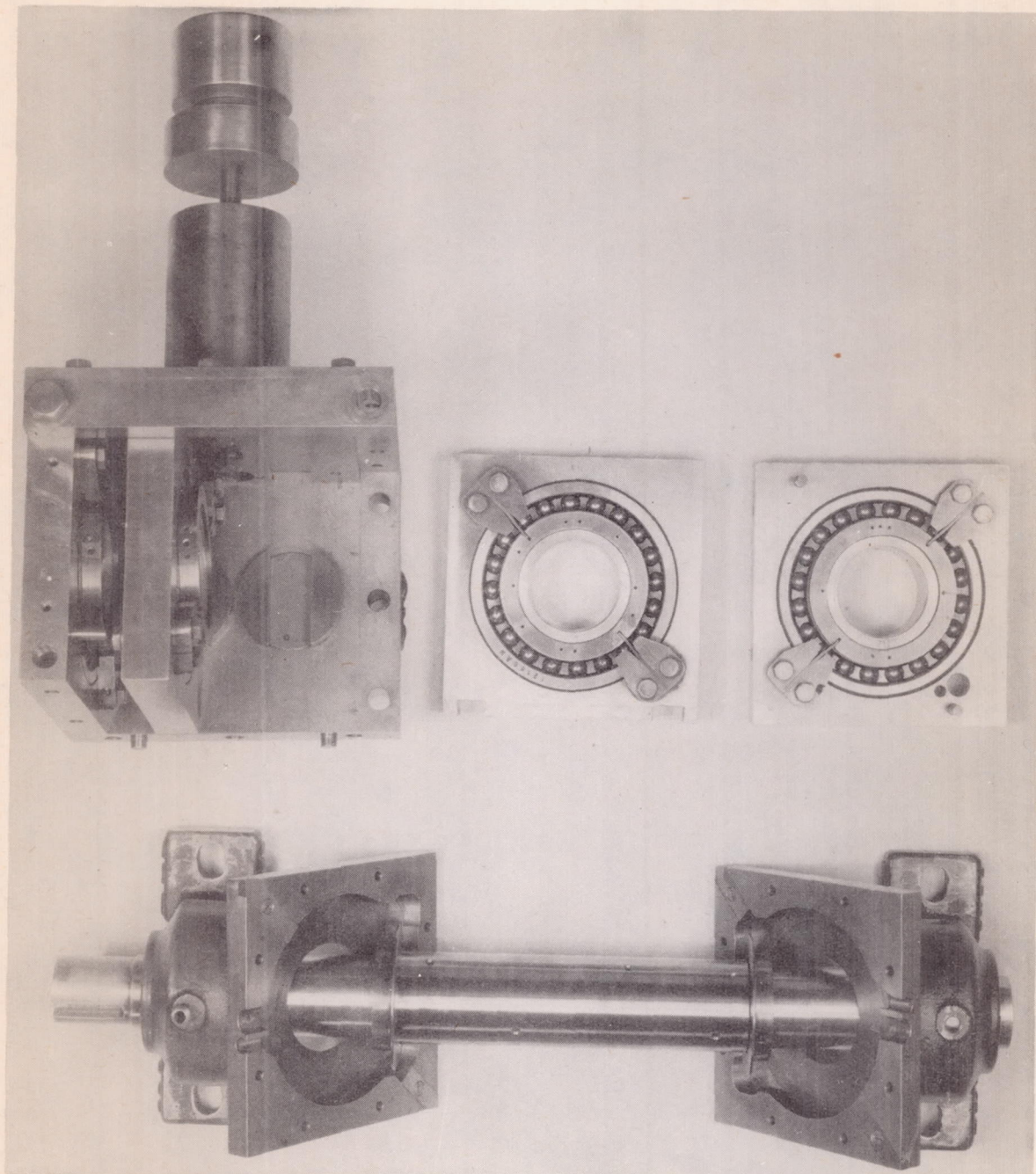


Figure 2.- Photograph of friction machine parts.



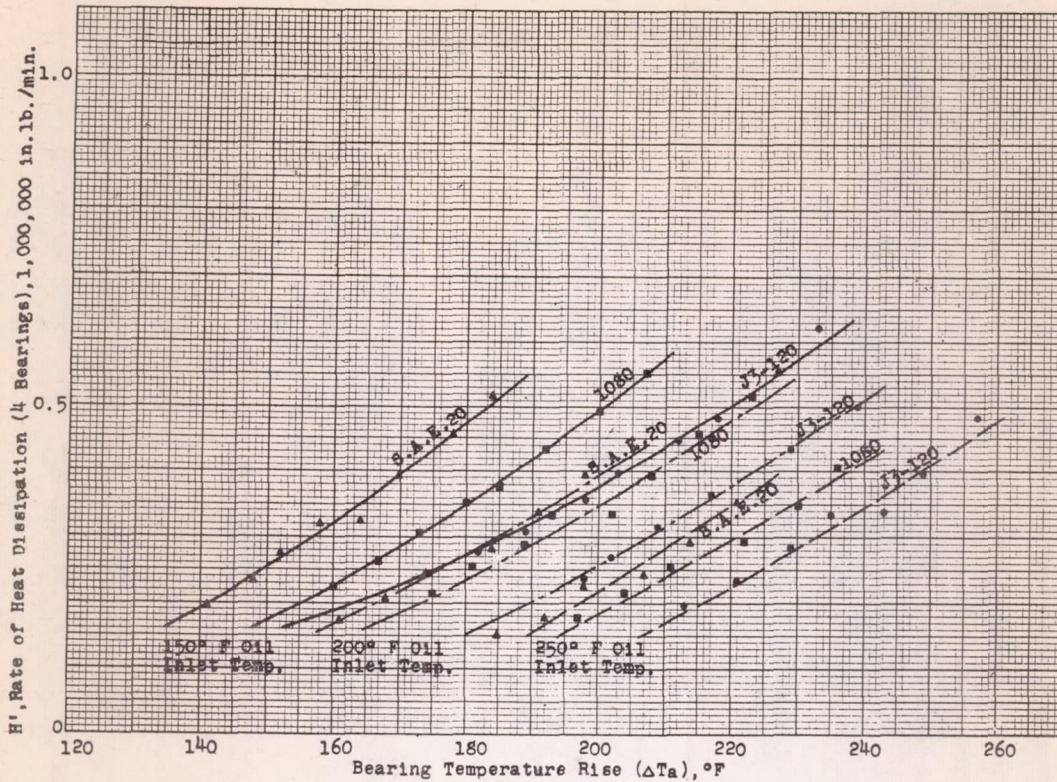


Figure 3.—Effect of viscosity grade and oil-inlet temperature upon the heat-dissipation characteristics of 2" x 1-1/4" copper-lead bearings, C/D = 0.0009.

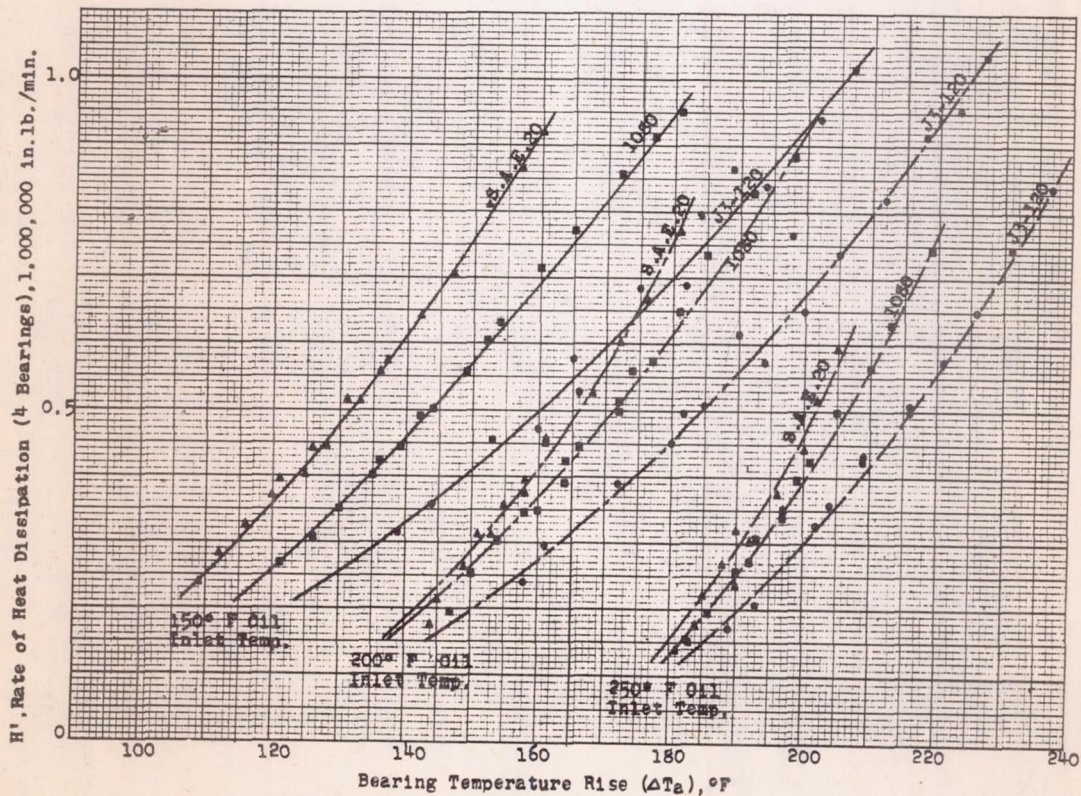


Figure 4.—Effect of viscosity grade and oil-inlet temperature upon the heat-dissipation characteristics of 2" x 1-1/4" copper-lead bearings, C/D = 0.0016.



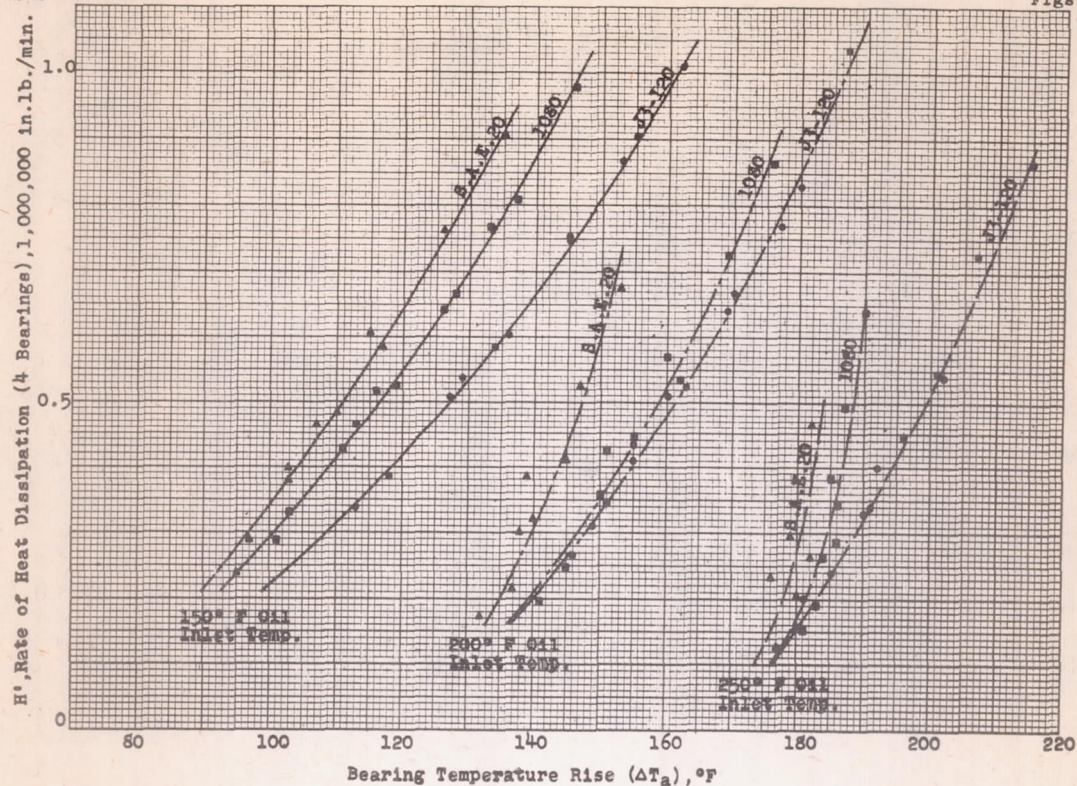


Figure 5.---Effect of viscosity grade and oil-inlet temperature upon the heat-dissipation characteristics of 2" x 1-1/4" copper-lead bearings, C/D = 0.0029.

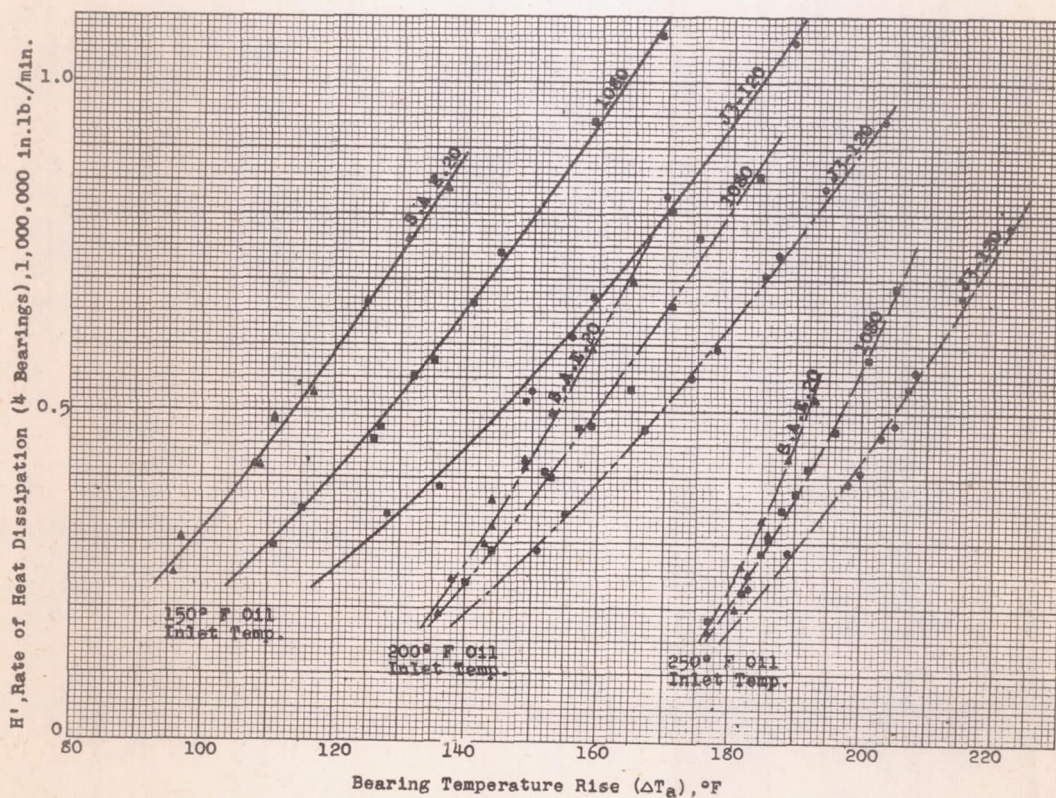


Figure 6.---Effect of viscosity grade and oil-inlet temperature upon the heat-dissipation characteristics of 2" x 1-1/4" lead-indium coated silver bearings, C/D = 0.0021.



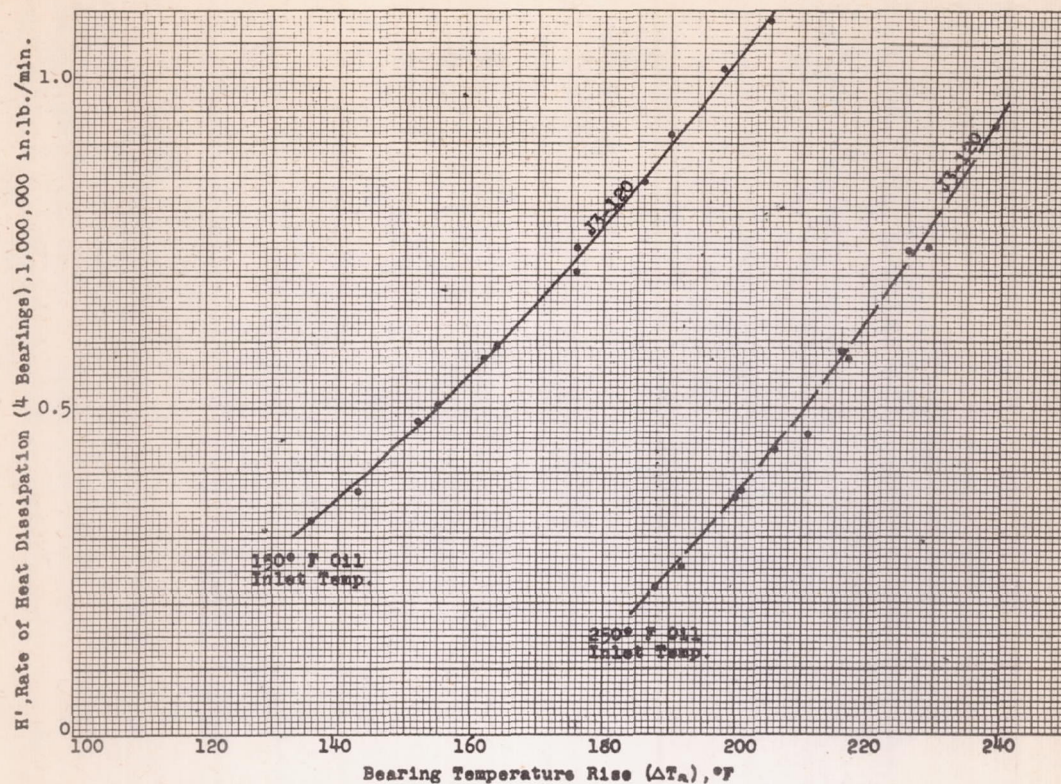


Figure 7.—Effect of oil-inlet temperature upon the heat-dissipation characteristics of 2" x 1-1/4" lead-coated copper-lead bearings,  $C/D = 0.0020$ .

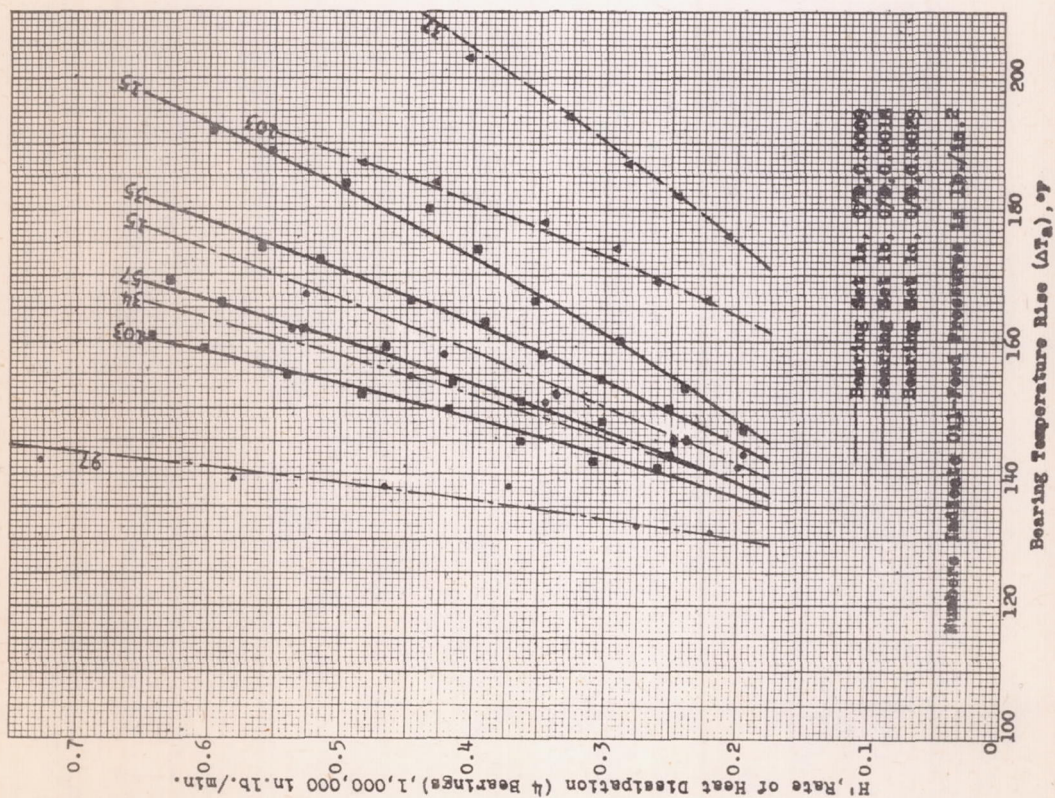


Figure 8.—Effect of oil-feed pressure upon the heat-dissipation characteristics of 2" x 1-1/4" copper-lead bearings.



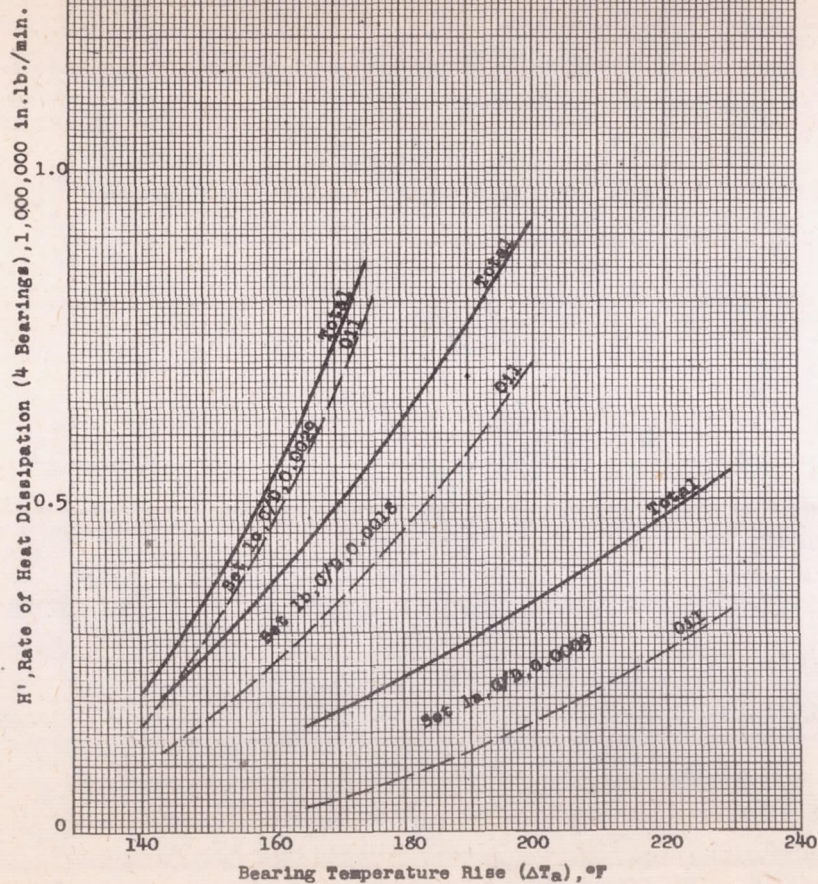


Figure 9.—Comparison between total heat dissipation and removal of heat by the oil flowing through the machine for 2" x 1-1/4" copper-lead bearings using Navy contract 1050 oil at 200°F oil-inlet temperature.

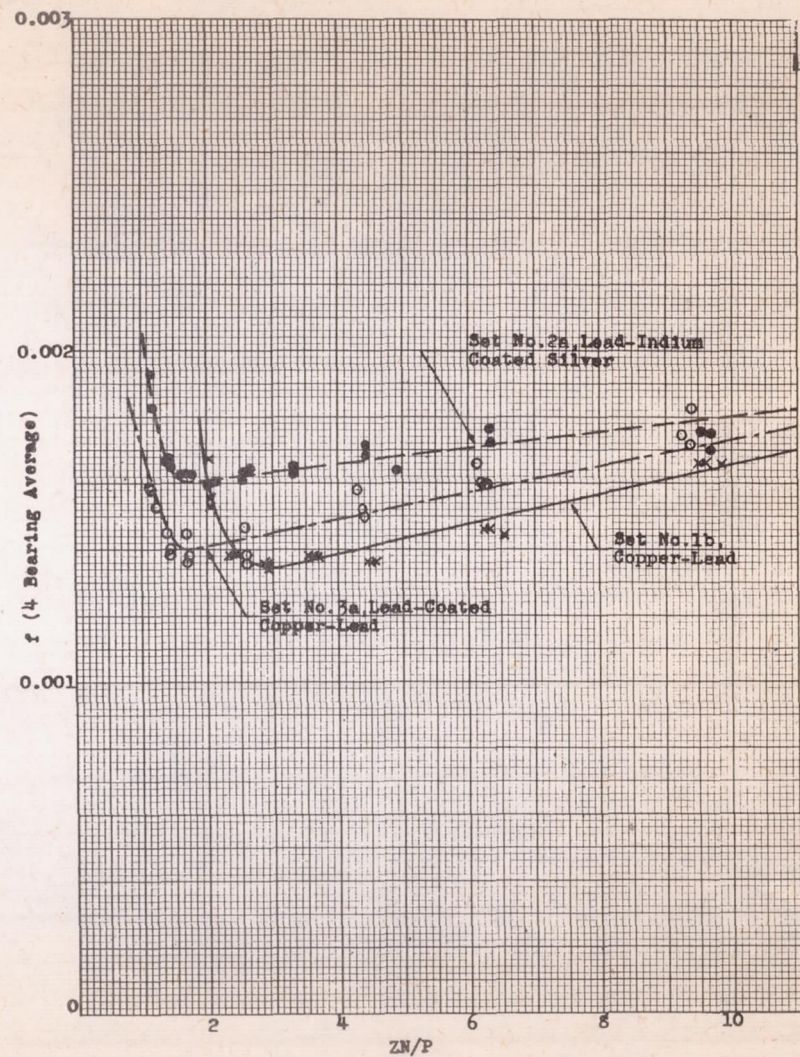


Figure 10.— $f$ - $ZN/P$  curves for 2" x 1-1/4" bearings of different metals.



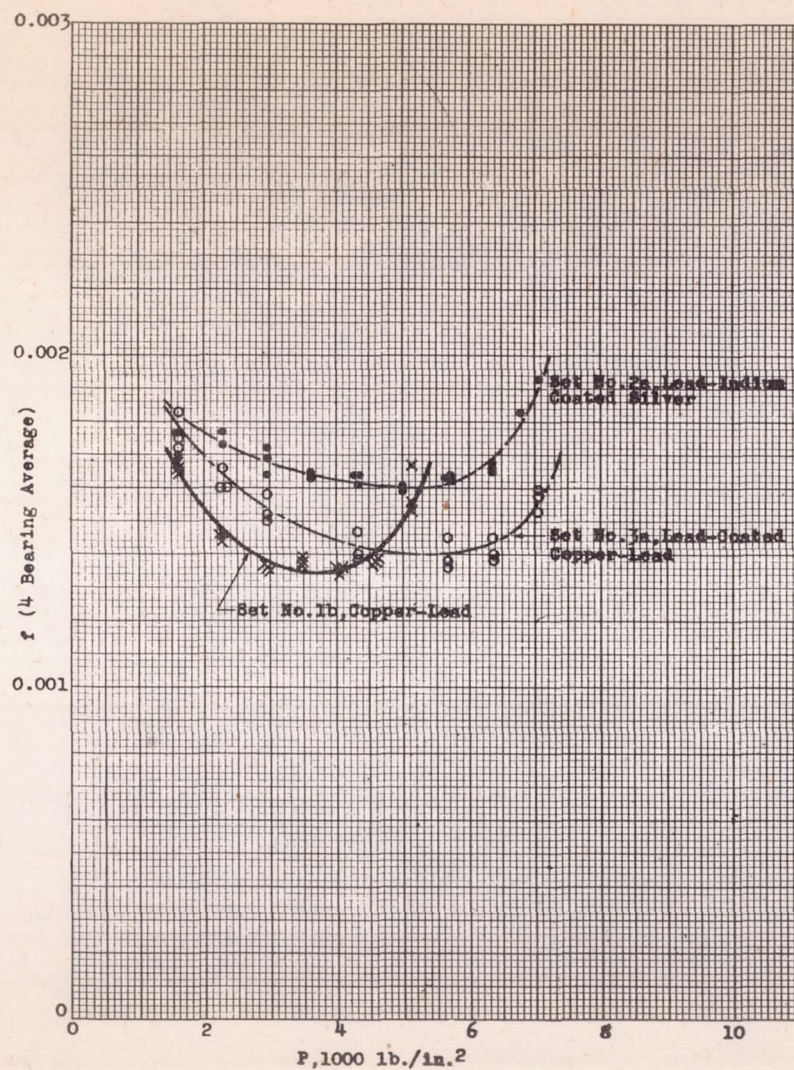


Figure 11.— $f$ - $P$  curves for 2" x 1-1/4" bearings of different metals.

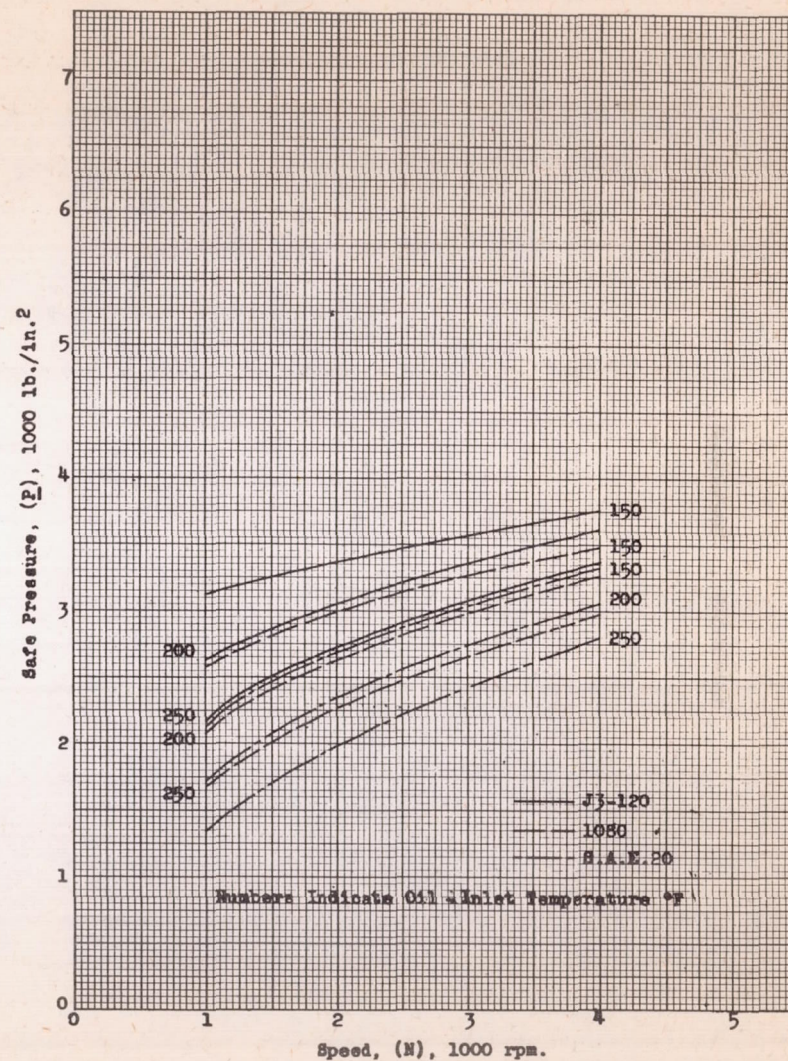


Figure 12.—Effect of viscosity grade and oil-inlet temperature upon the computed safe loads for 2" x 1-1/4" copper-lead bearings,  $C/D = 0.0009$ , at various speeds.



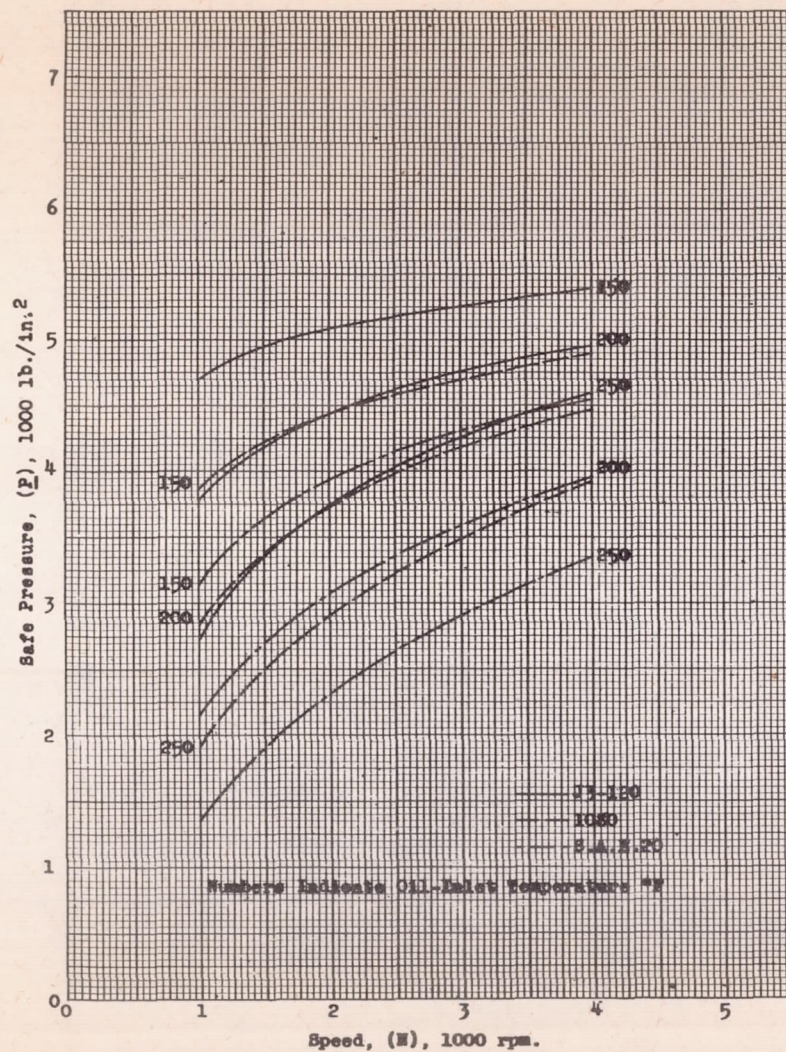


Figure 13.—Effect of viscosity grade and oil-inlet temperature upon the computed safe loads for 2" x 1-1/4" copper-lead bearings,  $C/D = 0.0018$ , at various speeds.

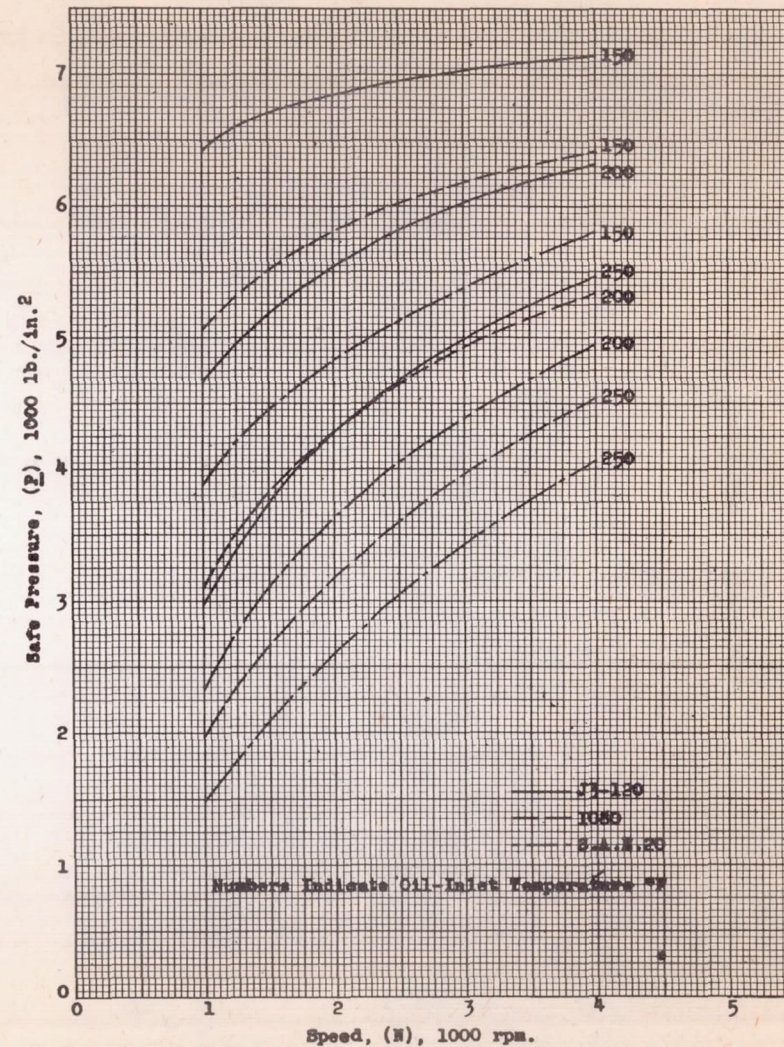


Figure 14.—Effect of viscosity grade and oil-inlet temperature upon the computed safe loads for 2" x 1-1/4" copper-lead bearings,  $C/D = 0.0029$ , at various speeds.



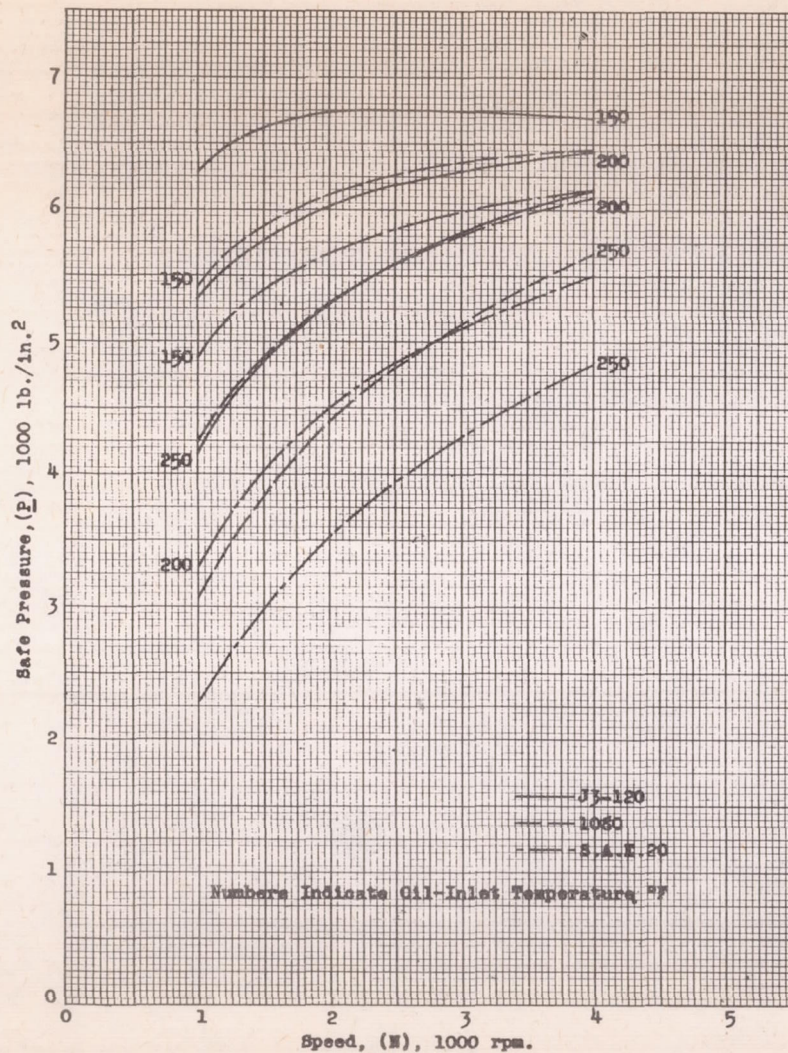


Figure 15.—Effect of viscosity grade and oil-inlet temperature upon the computed safe loads for 2" x 1-1/4" lead-indium coated silver bearings, C/D = 0.0018, at various speeds.

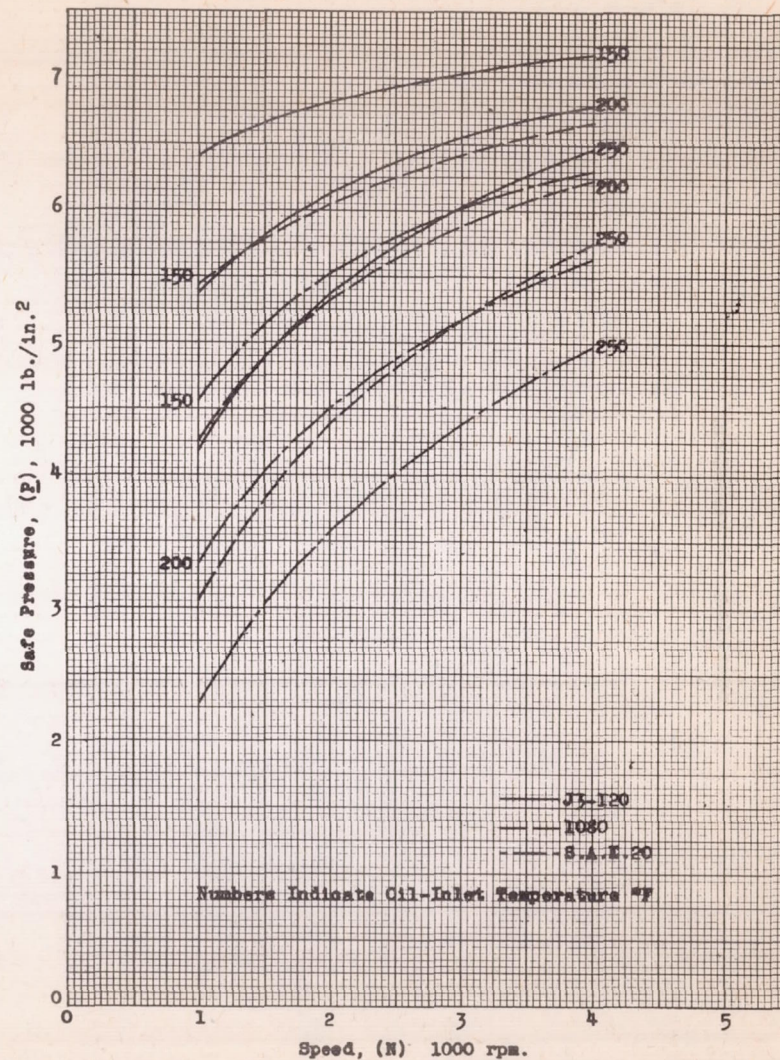


Figure 16.—Effect of viscosity grade and oil-inlet temperature upon the computed safe loads for 2" x 1-1/4" lead-coated copper bearings, C/D = 0.0018, at various speeds.



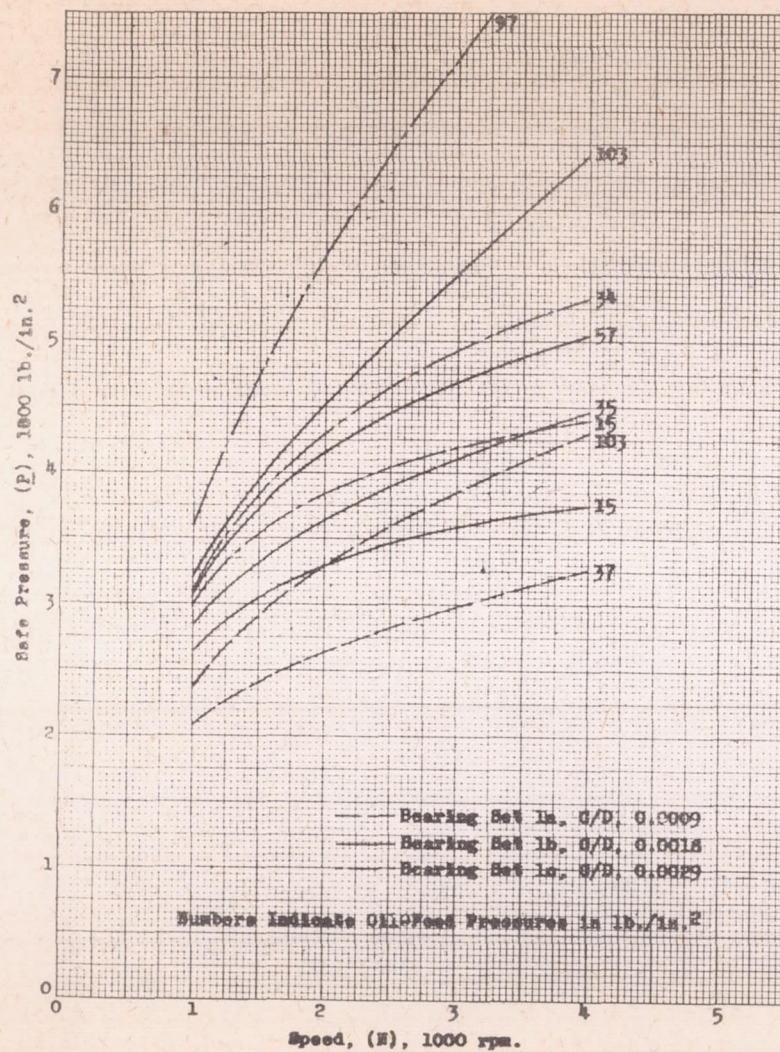


Figure 17.—Effect of oil-feed pressure upon the computed safe loads for 2" x 1-1/4" copper-lead bearings at various speeds.

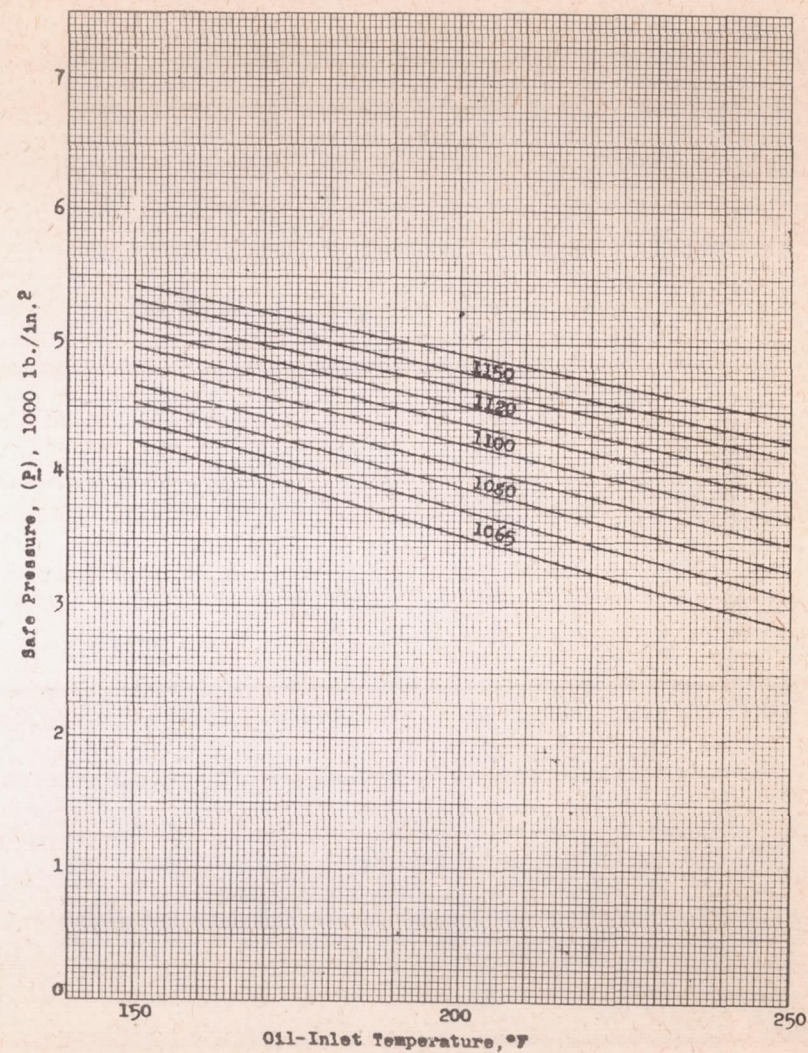


Figure 18.—Chart for determining safe loads for Army and Navy Specification Oils at various oil-inlet temperatures. 2" x 1 1/4" copper-lead bearings, C/D=0.0018, 2500 rpm.



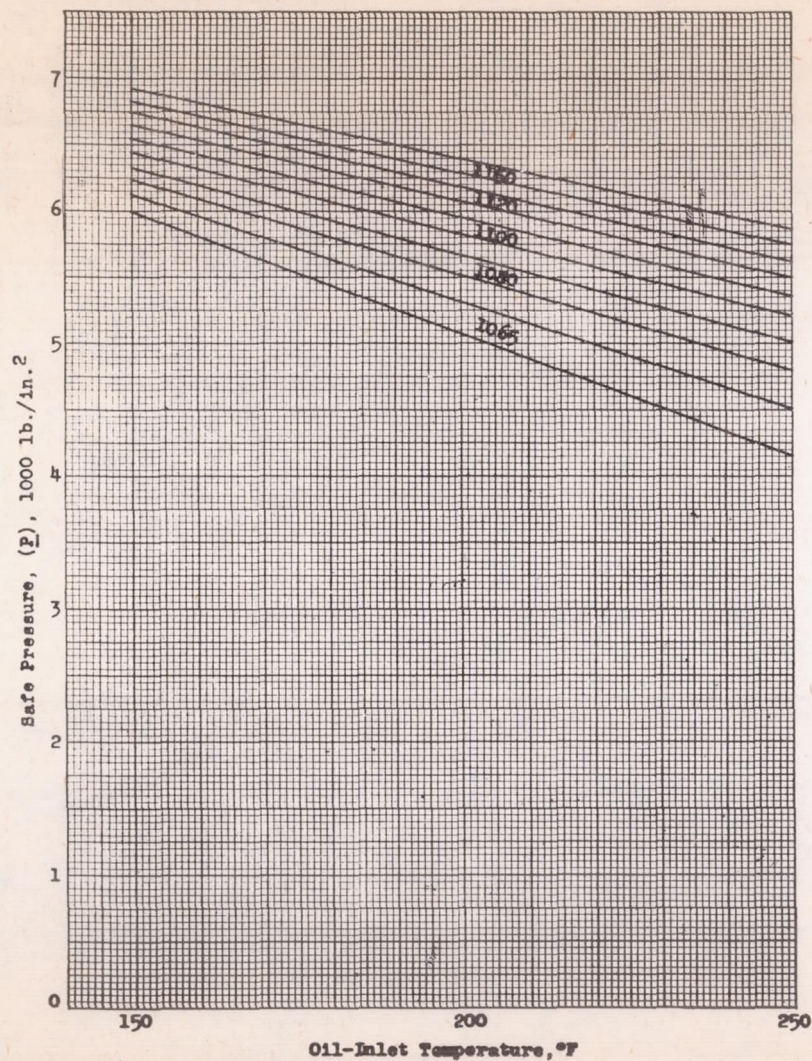


Figure 19.—Chart for determining safe loads for Army and Navy Specification Oils at various oil-inlet temperatures. 2" x 1 1/4" lead-indium coated silver bearings,  $C/D=0.0018$ , 2500 rpm.

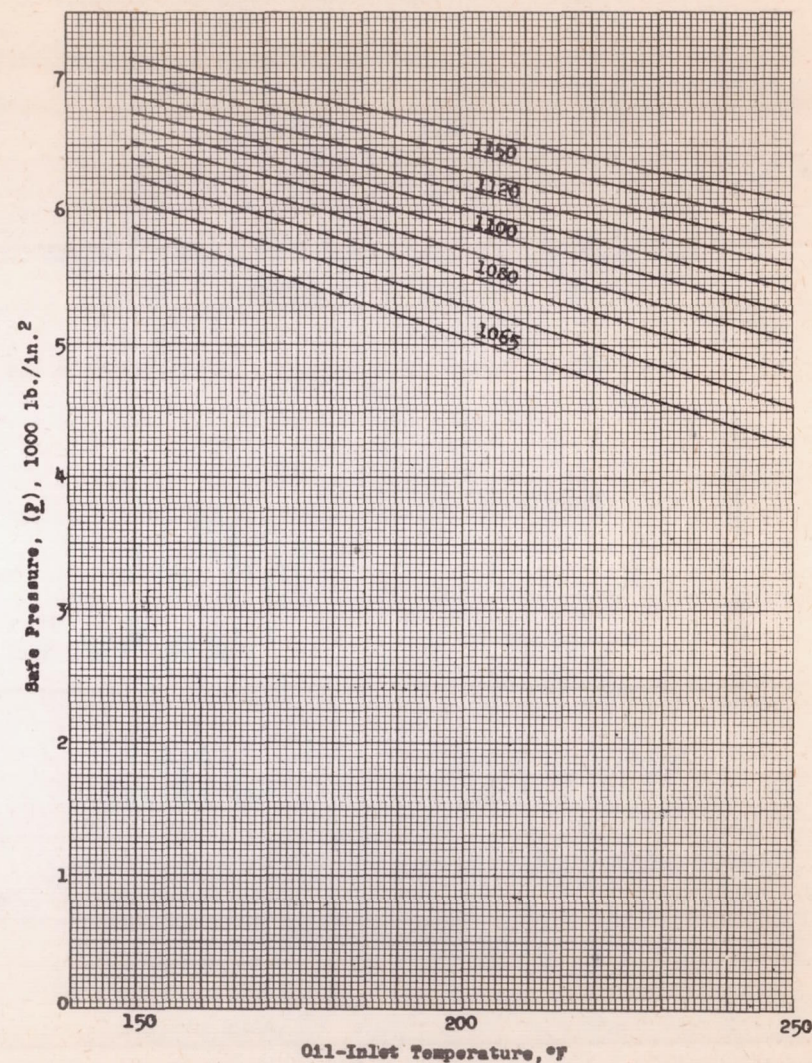


Figure 20.—Chart for determining safe loads for Army and Navy Specification Oils at various oil-inlet temperatures. 2" x 1 1/4" lead-coated copper-lead bearings,  $C/D=0.0018$ , 2500 rpm.



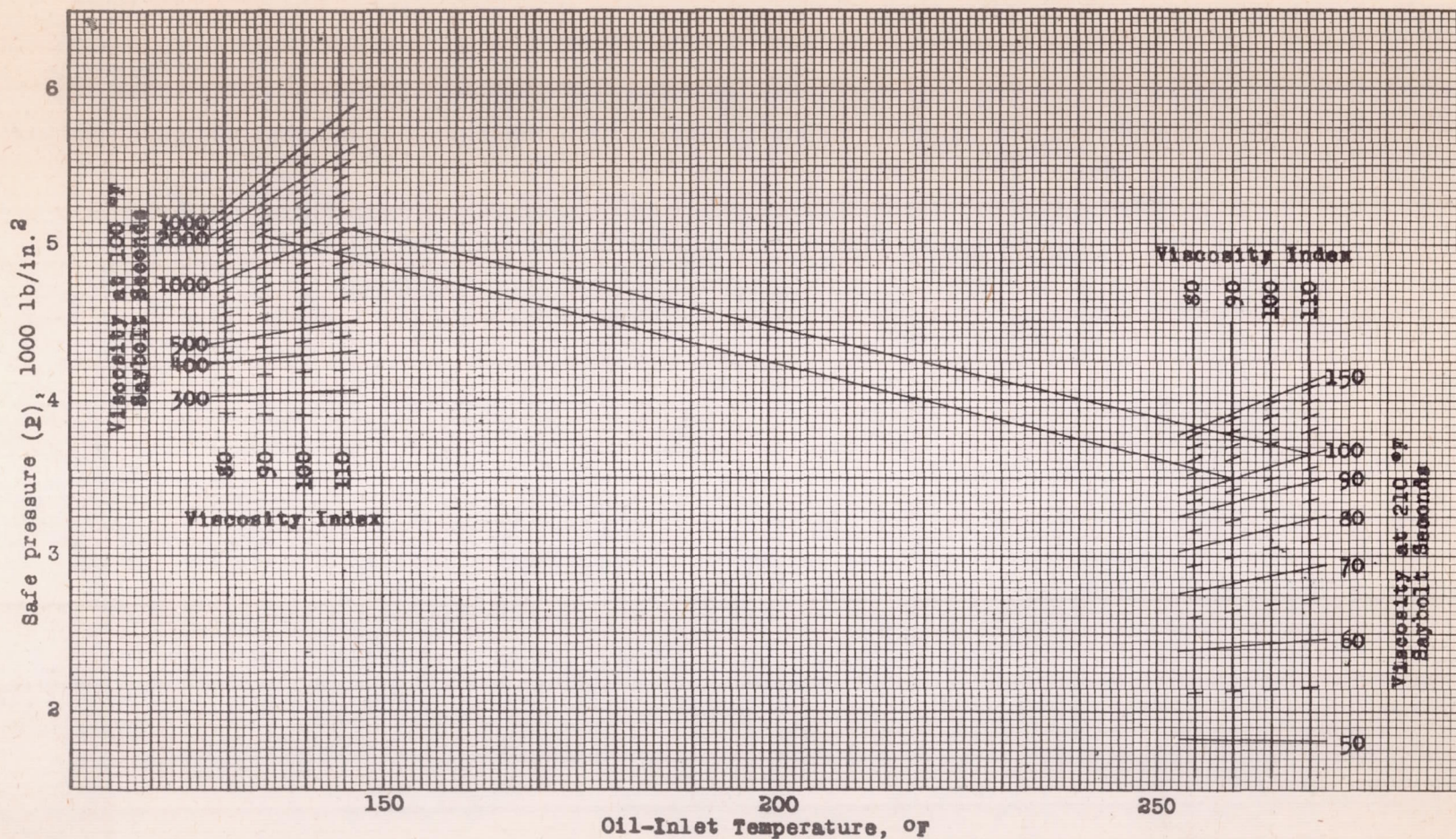


Figure 21.-- Chart for determining safe loads for oils of different viscosity grade and viscosity index at various oil-inlet temperatures. 3" xl-1/4" copper-lead bearings, C/D = 0.0018, 2500 rpm.

On the composite scale at the left of the chart, plot a point representing the viscosity of the oil at 100°F and the viscosity index of the oil. Plot a point representing the viscosity of the oil at 210°F and its viscosity index on the scale at the right of the chart. Draw a straight line between these two points. This line represents approximately the limits of safe pressures for this oil at the various oil-inlet temperatures.



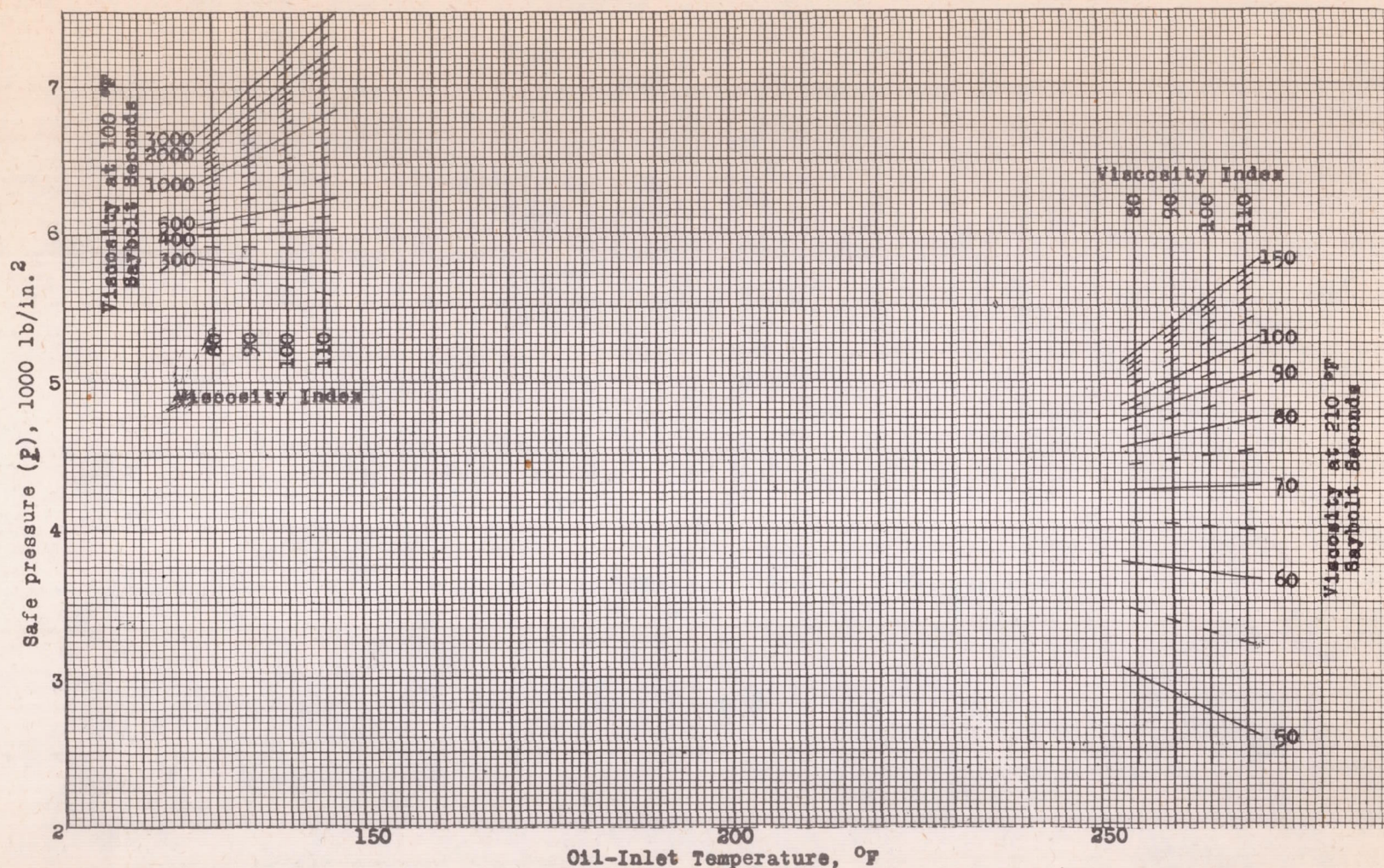


Figure 22.- Chart for determining safe loads for oils of different viscosity grade and viscosity index at various oil-inlet temperatures. 2" x 1-1/4" lead-indium coated silver bearings. C/D = 0.0018, 2500 rpm.

On the composite scale at the left of the chart, plot a point representing the viscosity of the oil at 100°F and the viscosity index of the oil. Plot a point representing the viscosity of the oil at 210°F and its viscosity index on the scale at the right of the chart. Draw a straight line between these two points. This line represents approximately the limits of safe pressures for this oil at the various oil-inlet temperatures.



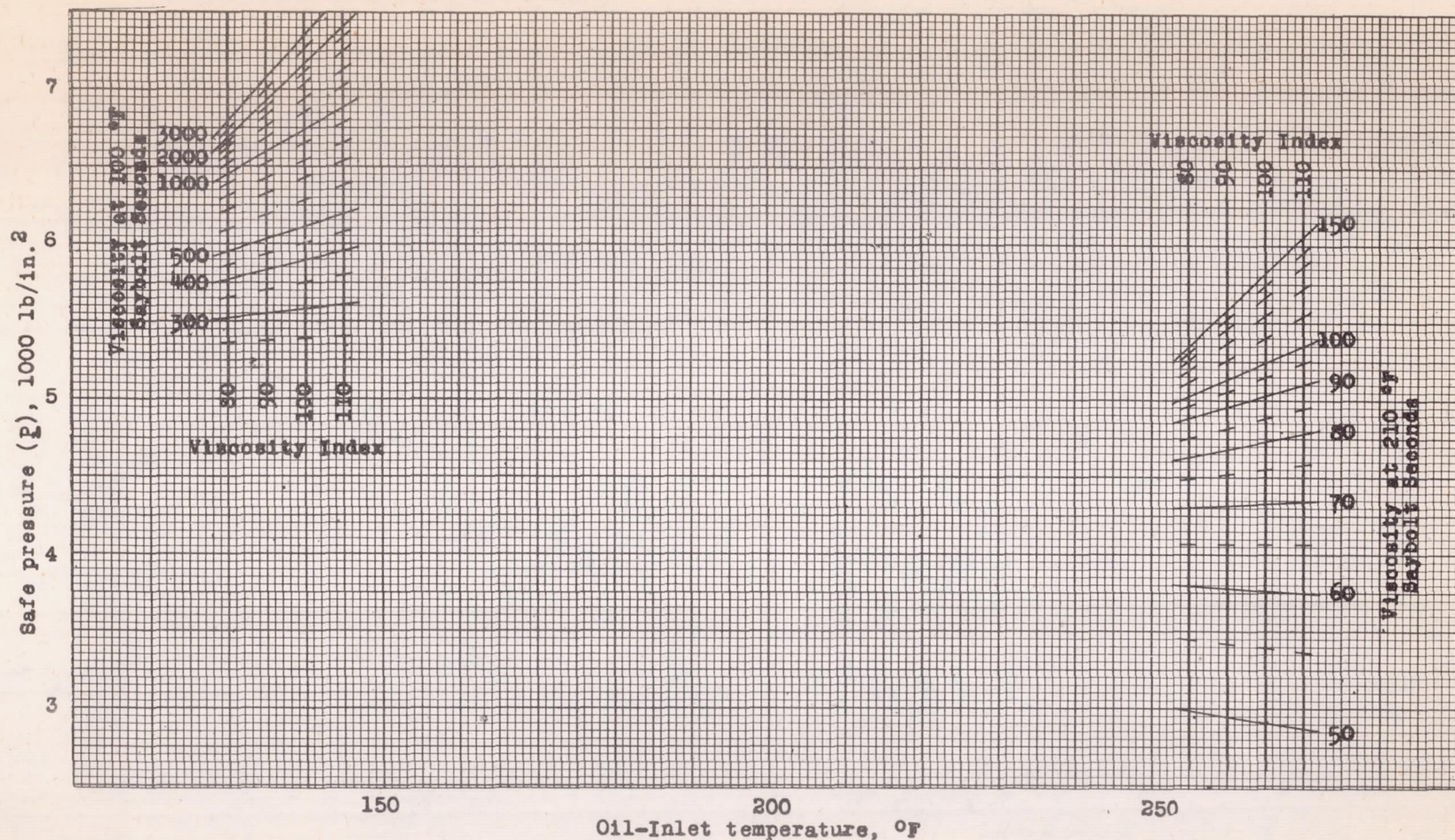


Figure 23.- Chart for determining safe loads for oils of different viscosity grade and viscosity index at various oil-inlet temperatures. 2" x 1-1/4" lead-coated copper-lead bearings, C/D = 0.0018, 2500 rpm.

On the composite scale at the left of the chart, plot a point representing the viscosity of the oil at 100°F and the viscosity index of the oil. Plot a point representing the viscosity of the oil at 210°F and its viscosity index on the scale at the right of the chart. Draw a straight line between these two points. This line represents approximately the limits of safe pressures for this oil at the various oil-inlet temperatures.



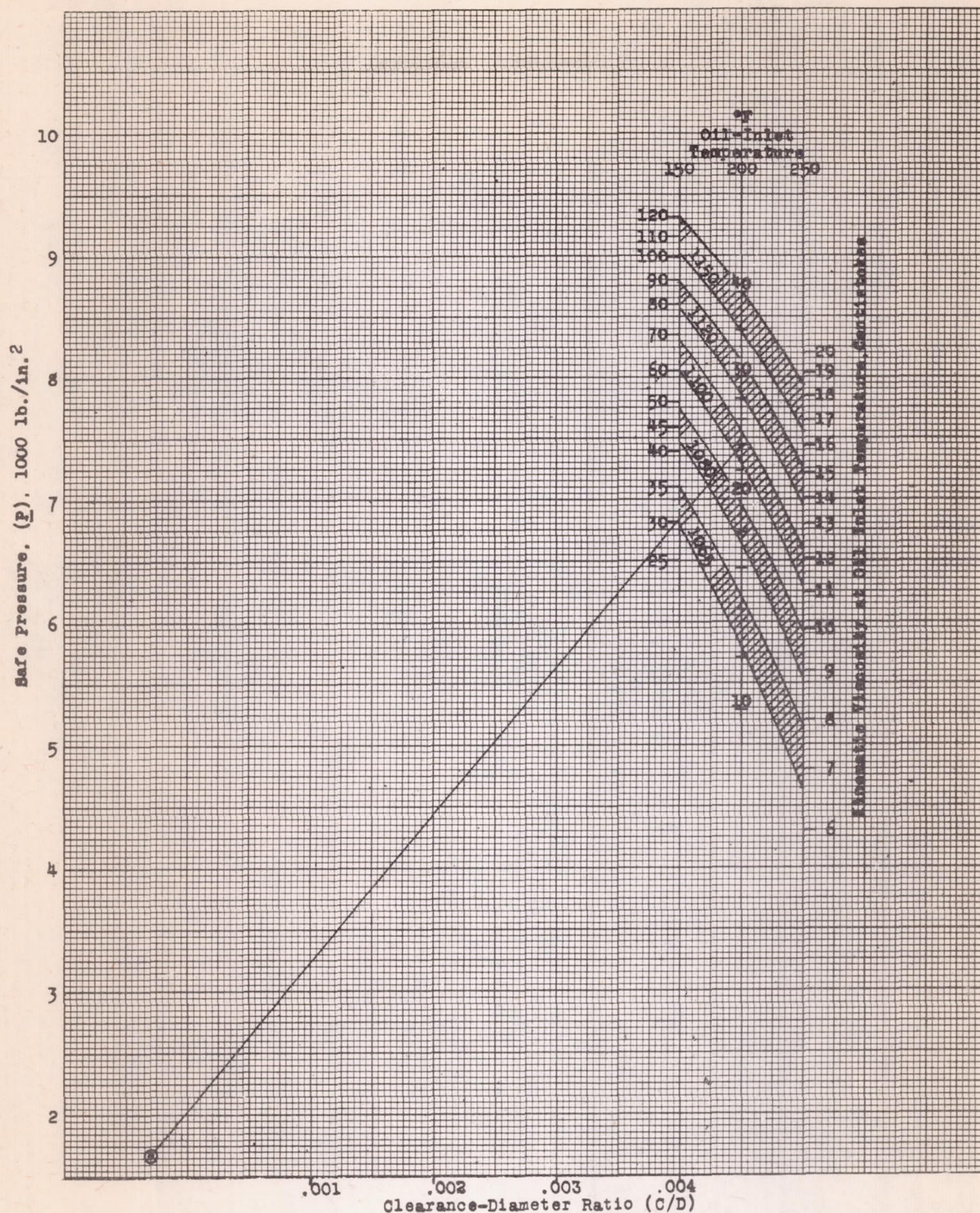


Figure 24.- Chart for determining safe loads for bearings of different clearance using oils of various viscosity grades at various oil-inlet temperatures. 2" x 1-1/4" copper-lead bearings, 2500 rpm.

Plot the viscosity of the oil at the oil-inlet temperature under consideration. Draw a straight line between this point and the bull's-eye at the lower left of the chart. This line defines approximately the limiting safe pressures for the clearance-diameter ratios when using the given oil, oil-inlet temperature, and speed.



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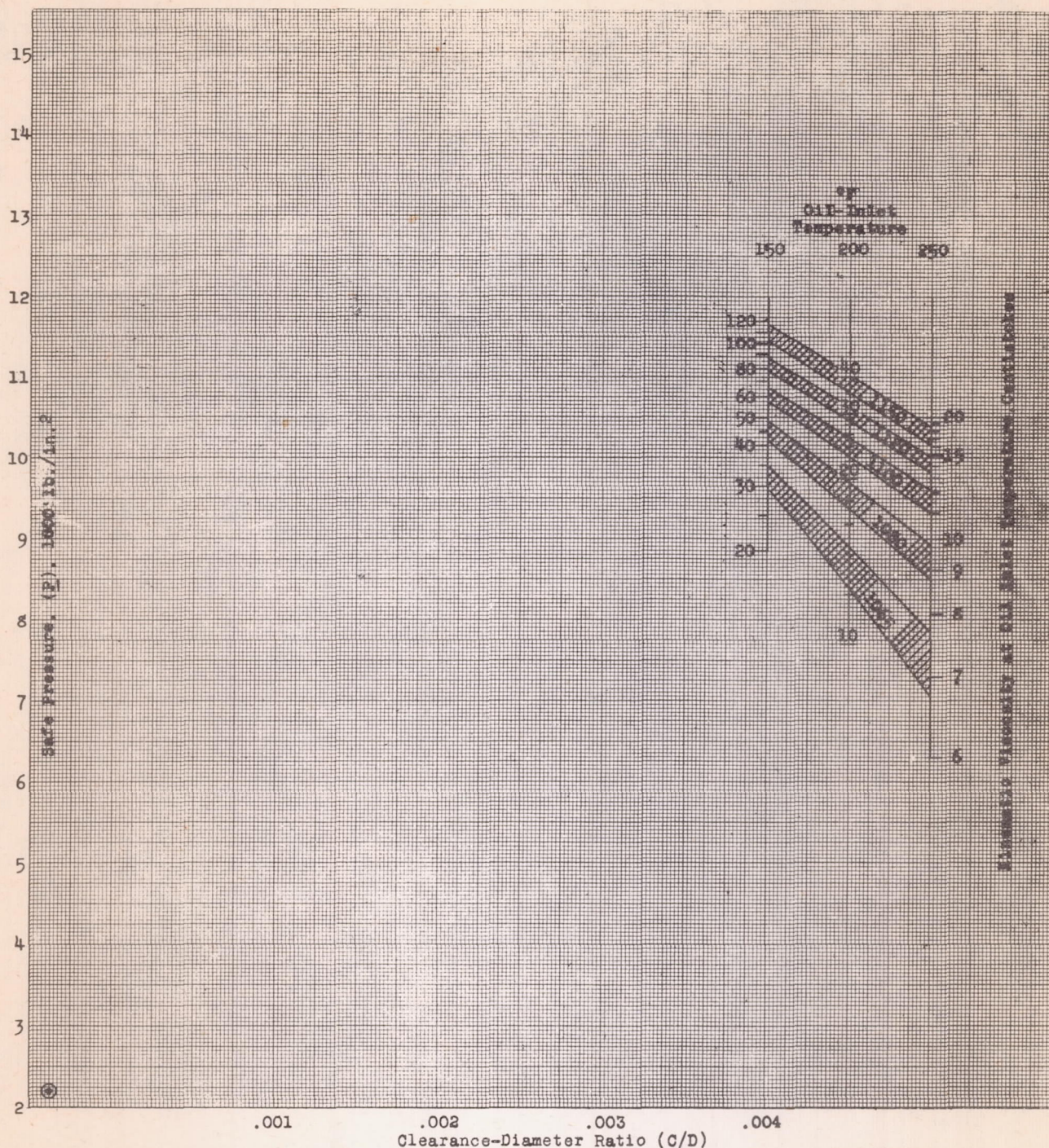


Figure 25.- Chart for determining safe loads for bearings of different clearance using oils of various viscosity grades at various oil-inlet temperatures. 2" x 1-1/4" lead-indium coated silver bearings, 2500 rpm.

Plot the viscosity of the oil at the oil-inlet temperature under consideration. Draw a straight line between this point and the bull's-eye at the lower left of the chart. This line defines approximately the limiting safe pressures for the clearance-diameter ratios when using the given oil, oil-inlet temperature, and speed.



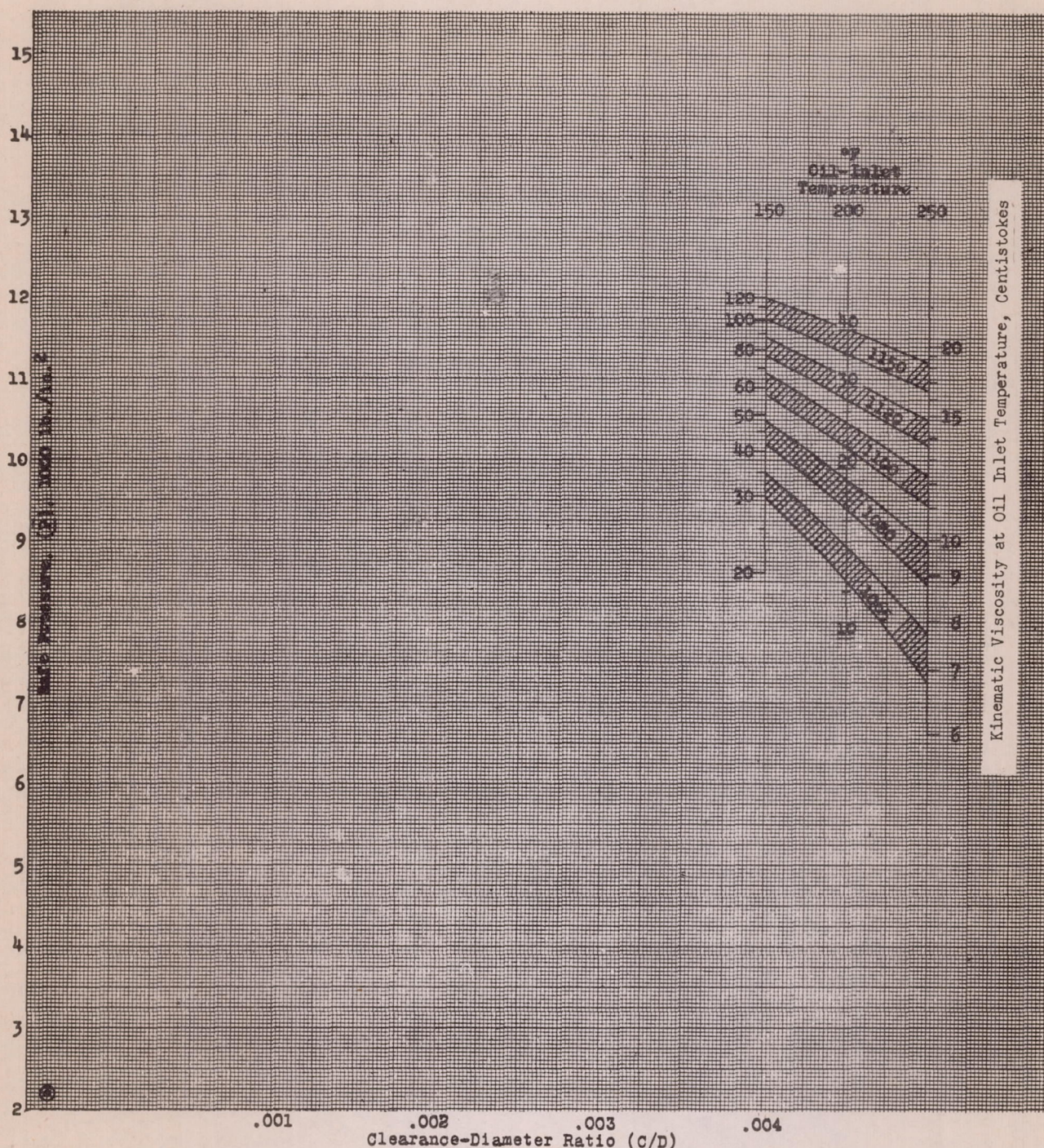


Figure 26.- Chart for determining safe loads for bearings of different clearance using oils of various viscosity grades at various oil-inlet temperatures. 2" x 1-1/4" lead-coated copper-lead bearings, 2500 rpm.

Plot the viscosity of the oil at the oil-inlet temperature under consideration. Draw a straight line between this point and the bull's-eye at the lower left of the chart. This line defines approximately the limiting safe pressures for the clearance-diameter ratios when using the given oil, oil-inlet temperature, and speed.